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COGENERATION TECHNOLOGY ALTERNATIVES STUDY (CTAS) UNITED TECHNOLOGIES CORPORATION FINAL REPORT

VOLUME III – ENERGY CONVERSION SYSTEM CHARACTERISTICS

Power Systems Division
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VOLUME III

PREFACE

The Cogeneration Technology Alternatives Study (CTAS) was preformed by the National Aeronautics and Space Administration, Lewis Research Center, for the Department of Energy, Division of Fossil Fuel Utilization. CTAS is aimed at providing a data base which will assist the Department of Energy in establishing research and development funding priorities and emphasis in the area of advanced energy conversion system technology for advanced industrial cogeneration applications. CTAS includes two Department of Energy-sponsored/National Aeronautics and Space Administration-contracted studies conducted in parallel by industrial teams along with analyses and evaluations by the National Aeronautics and Space Administration's Lewis Research Center.

This document describes the work conducted by Power Systems Division of United Technologies Corporation under National Aeronautics and Space Adminstration contract DEN3-30. This United Technologies contractor report is one of a set of reports describing CTAS results. The other reports are the following: Cogeneration Technology Alternatives Study (CTAS) Volume I - Summary NASA TM 81400, Cogeneration Technology Alternatives Study (CTAS) General Electric Final Report NASA CR 159765-159770 and Cogeneration Technology Alternatives Studies (CTAS) Volume II - Comparison and Evaluation of Results, NASA TM 81401.

This United Technologies contractor report for the CTAS study is contained in six volumes:

Volume I	- Summary Report, DOE/NASA/0030-80/1 NASA CR 159759
Volume II	- Industrial Process Characteristics, DOE/NASA/0030-80/2 NASA CR 159760
Volume III	- Energy Conversion System Characteristics, DOE/NASA/0030-80/3 NASA CR 159761
Volume IV	- Heat Sources, Balance of Plant, and Auxiliary Systems, DOE/NASA/0030-80/4 159762
Volume V	- Analytic Approach and Results, DOE/NASA/0030-80/5 159763
Volume VI	- Computer Data, DOE/NASA/0030-80/6 NASA CR 159764

The following persons or organizations have developed and provided data and information for the current and advanced energy conversion system technologies used in this Cogeneration Technology Alternatives Study and presented in this Volume III:

DeLaval Turbine and Compressor Division of Trenton, New Jersey
Cummins Cogeneration Company of New York, New York

Cummins Engine Company of Columbus, Indiana
Dr. P. S. Myers of Madison, Wisconsin
Sulzer Brothers, Limited, of Winterthur, Switzerland
Mechanical Technology Incorporated of Latham, New York
Rasor Associates, Incorporated of Sunnyvale, California
Aerojet Energy Conversion Company of Sacramento, California
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VOLUME III

INTRODUCTION

The Cogeneration Technology Alternatives Study evaluated the advantages of advanced energy conversion technologies in industrial cogeneration applications. To meet the objectives of this study, data and information had to be established for: (1) both current and future energy conversion technologies, (2) representative process plants in energy intensive industries, (3) heat sources (or furnaces) as required by the conversion systems, (4) balance of plant, (5) supporting technologies, and (6) study assumptions and ground rules. These data were analyzed and conservation, economic, and environmental impact were evaluated at the industrial plant level and extrapolated to the potential national level for each energy conversion system-fuel combination.

The study in its entirety is summarized in Volume I of this report. A series of five additional volumes provide the more detailed data and information which was used in the course of the study. The purpose of this Volume III is to present current and advanced technology energy conversion system characterizations provided by specialists in each technology.

The conversion technologies were selected for applicability in industrial processes. Front end, or topping, systems which produce electricity and recover heat for the process were emphasized. Back-end, or bottoming, configurations which receive thermal energy from the industrial process and produce electricity for use at the plant site or for export to the electric utility were considered. The conversion systems were also chosen with emphasis on the potential for energy savings and transition from the use of oil and natural gas to coal or coal-derived fuels. The advanced technologies were selected to be consistent with technology estimated to be commercially available in the 1985-2000 period. In making this judgment, the existence of, or lack of, advanced technology programs or development programs supported by either industry or government was not considered.

The selected current and advanced technology energy conversion systems and associated fuels are presented in Table III-1. Five cogeneration technologies, steam turbines, gas turbines, combined cycle and two classes of diesel engines are representative of current technology and use conventional fuels, oil and gas. The advanced energy conversion systems include steam turbines; gas turbines, both direct and indirect fired; combined cycle; closed (or Brayton cycle) gas turbines; steam injected gas turbines; two classes of diesel engines; thermionic conversion; low and high temperature fuel cells; stirling engines; and, for bottoming cogeneration applications only, organic Rankine cycle. The fuels selected were generally the heaviest, least refined type appropriate for the technology in the 1985-2000 time period. Coal or coal-derived fuels were selected where possible.

The industrial process energy requirements in this study vary over a wide range. Some require a large amount of low temperature heat (usually hot water or low pressure steam) and others require substantial amounts of intermediate or high temperature heat. The choice of energy conversion system design conditions can emphasize heat recovery at one temperature or another. For purposes of this study, the industrial thermal requirements have been classified in five categories: hot water, steam at three temperatures, and hot gases. To provide the greatest applicability for each technology, a number of design configurations emphasizing performance in one or more of these categories were chosen by the technical specialists. In like manner, the electrical to thermal energy requirements ratio varies from industrial process to process. The technical advocate for each technology recognized this variability and provided data and information for designs calculated to produce the greatest cogeneration benefits. Up to five design alternatives were considered for each of the energy conversion systems identified in Table III-1.

TABLE III - 1

ENERGY CONVERSION SYSTEMS

Number	Type		Fuel
CURRENT TECHNOLOGY - TOPPING APPLICATIONS			
1	Steam Turbine		Petroleum, Boiler Grade
2	Steam Turbine		Coal (FGD)
3	Diesel	High Speed	Petroleum Distillate
4	Diesel	Low Speed	Petroleum Boiler Grade
5	Gas Turbine	Direct	Petroleum Distillate
6	Combined Cycle	Direct	Petroleum Distillate
ADVANCED TECHNOLOGY - TOPPING APPLICATION			
7	Steam Turbine		Coal Derived Boiler Grade
8	Steam Turbine		Coal (AFB)
9	Diesel,	High Speed	Coal Derived Distillate
10	Diesel,	Low Speed	Coal Derived Boiler Grade
11	Diesel	Low Speed	Coal (pulverized)
12	Gas Turbine,	Direct Fired	Petroleum Boiler Grade
13	Gas Turbine	Direct Fired	Coal Derived Boiler Grade
14	Gas Turbine	Direct Fired	Coal Derived Low BTU Gas
15	Gas Turbine	Direct Fired	Coal (PFB)
16	Gas Turbine,	Indirect	Coal (AFB)
17	Gas Turbine	Closed Cycle	Coal Derived Boiler Grade
18	Gas Turbine	Closed Cycle	Coal (AFB)
19	Steam Injection Gas Turbine,	Direct Fired	Petroleum Boiler Grade
20	Steam Injection Gas Turbine,	Direct Fired	Coal Derived Boiler Grade
21	Steam Injection Gas Turbine,	Direct Fired	Coal (PFB)
22	Steam Injection Gas Turbine	Indirect	Coal (AFB)
23	Combined Cycle	Direct	Petroleum Boiler Grade
24	Combined Cycle	Direct	Coal Derived Boiler
25	Combined Cycle	Direct	Coal (PFB)
26	Combined Cycle	Indirect	Coal (AFB)
27	Fuel Cell	Low Temp	Petroleum Distillate
28	Fuel Cell	Low Temp	Coal Derived Distillate
29	Fuel Cell	High Temp	Petroleum Distillate
30	Fuel Cell	High Temp	Coal Derived Distillate
31	Fuel Cell	High Temp	Coal (Gasifier)
32	Stirling		Coal Derived Boiler Grade
33	Stirling		Coal (AFB)
34	Thermionics		Coal Derived Boiler Grade
35	Thermionics Compound		Coal Derived Boiler Grade
CURRENT TECHNOLOGY - BOTTOMING APPLICATIONS			
36	Steam Turbine		By-Product Heat
ADVANCED TECHNOLOGY - BOTTOMING APPLICATIONS			
37	Organic Rankine		By-Product Heat

For each design option, the energy conversion system performance was defined in terms of electrical output and heat recovery in each of the thermal categories. These data were determined over an applicable size range. Off-design performance was also determined. Performance data were related to the higher heating value of the fuel to determine efficiency or fuel utilization. In addition, equipment and installation costs were estimated. Also, maintenance frequency and cost were predicted. The exhaust emissions were determined and the physical size and weight of the energy conversion systems were defined.

This volume is arranged by groups of energy conversion systems: steam turbines, diesel engines, gas turbines, combined cycles, fuel cells, stirling engines, thermionics and organic Rankine cycles. It includes for each of these energy conversion systems a description of the system, performance data, cost estimates, predicted emissions, physical data, discussion of cogeneration applicability and technical developments to achieve commercialization in the 1985-2000 time period. Heat source data, including cost data, are presented in Volume IV.

Printouts of the energy conversion system data used in the computer computations are presented in Volume VI, Table VI-10, for the various design options evaluated in the study.

STEAM TURBINES

INTRODUCTION

Historically, the steam turbine has been the principal cogeneration energy conversion technology. It is able to provide both shaft power for mechanical drive and electric generation and steam for industrial processes. The Annual Survey of Manufacturers 1976, indicates that about 10 percent of industrial electrical energy consumption is generated at the industrial facility. The paper, petroleum, chemical, and metal industries generate significant amounts of electricity, and most of this power is produced by steam turbines. Almost any thermal and electrical requirement can be satisfied with sufficient steam pressure drop, particularly if process steam and shaft power requirements coincide.

The simplest steam turbine arrangement is a non-condensing or back pressure configuration where the turbine produces the required shaft power, and the exhaust steam provides the process thermal needs. This configuration has certain drawbacks, however. If the turbine drives an electric generator, variations in either process steam requirements or electrical demand can lead to complications. Also, the back pressure turbine has limited electrical-to-thermal energy ratios available. If the power requirements exceed the turbine output for a particular thermal need, two alternatives are evident. In one, needed additional electricity is purchased from the utility (a match thermal strategy). The other approach is to use an automatic extraction turbine with a condensing exhaust.

Automatic extraction units bleed part of the main steam flow at one, two or even three locations within the turbine. Valves in chambers between selected turbine stages control extracted steam pressure at the required level. The amount of steam extracted for process use at various points may be controlled in almost any manner within design limitations. When the extracted steam flowing through the turbine does not provide sufficient shaft power to meet the requirement, more steam can be introduced which flows through to the exhaust condenser. Automatic governing systems provide control for shaft speed and steam flows and pressures.

The basic design of the single extraction condensing turbine used in this study is shown in Figure III-1. The multi-stage steam turbine is of the impulse type and consists of a case containing a series of wheels each running in a separate compartment. The impulse type design results in minimum axial thrust upon the rotor. Therefore, steam economy is not dependent upon the maintenance of close clearances.

The turbine casing is split horizontally in the plane of the shaft center line. The turbine is provided with down exhaust to permit mounting of the turbine generator above grade. The steam chest is made of steel and is designed to accommodate rapid transients. The diaphragms between turbine disks are located in grooves, machined in the casing and are split horizontally. The nozzle vanes are welded to inner and outer rings to form a squirrel cage type structure. The rotor and wheels are machined from a single alloy steel forging. Solid rotor construction permits rapid warmup and startup of these turbines and is not dependent upon shrink fits between wheels and shafts. Rotating blades are machined from bar-stock or rough forged sections. The bases of the blades are machined to permit proper assembly in the turbine wheels. Typical configurations are locked "T", external dovetail, or bulb and shank. To prevent blade errosion due to moisture, stellite inserts are applied to the leading edges of some blades.

To prevent steam leakage from stage to stage, each diaphragm is provided with a labyrinth type seal. Also, where a shaft passes through the turbine casing, seals are provided to prevent leakage of steam. A system of piping is provided so that the leakage from the high pressure packing will seal the low pressure packing at normal conditions.

The turbine rotor is supported by the two tilting shoe, segmented, self-aligning, journal bearings. A Kingsbury double-acting thrust bearing of the segmental type is provided in order to maintain the axial position of the rotor.

A combined trip and throttle valve is furnished for the high pressure inlet of the turbine. It functions as a closing valve if the overspeed governor should trip.

The flow of steam to the front end turbine is controlled through a series of vertical bar lifts which automatically admit steam to different nozzle groups.

The shaft drives a synchronous, three-phase alternator at 3600 RPM. For small size turbines a gear box is introduced to permit higher turbine operating speed while maintaining the generator at 3600 RPM.

The turbine-generator requires a steam supply, condenser, and heat rejection system to function as a complete energy conversion system. The characteristics of the applicable steam generators are presented in Volume IV, Heat Sources, Heat Storage, and Balance of Plant. That volume includes the necessary balance-of-plant items for a complete system. They include the fuel storage and distribution system, the limestone storage and distribution system for the coal combustion systems, waste disposal systems, the sulfur dioxide scrubber system (as applicable), the boiler feedwater system, the heat rejection system, the electrical conditioning and control system, site preparation, and installation. The characteristics of the complete energy conversion system are reported in this volume. For detail information concerning the heat source and balance-of-plant, Volume IV should be consulted.

CONVERSION SYSTEM DESCRIPTION

The principle elements of a complete steam turbine energy conversion system are included in Figures III-2, III-3, or III-4. Either an oil-fired or coal-fired heat source produces the steam to operate the turbine and to be extracted for the process. With boiler oil fuel (or coal-derived boiler fuel), the system is designed to meet the pollution ground rules. In like manner, the coal-fired heat sources incorporate flue gas desulfurization or fluid bed combustion to help keep pollutants within the requirements.

Large utility turbines have been designed for steam pressures as high as 4500 psi and temperatures up to 1200°F, but current industrial practice typically uses pressures up to 1200 or 1400 psi and temperatures up to 950°F. These levels have

evolved over a period of years and reflect established design practice and materials. Therefore, inlet steam pressure of 1200 psig and temperature of 950°F were selected for the current technology steam turbines in this study. Extraction pressures of 600 and 50 psig are consistent with the thermal bin system selected for this study. Condenser pressure of 3 psia was determined by the design of the evaporative, mechanical draft, cooling tower, Volume IV, and a minimum temperature difference of 30°F.

In the 1985 - 2000 period industrial steam turbine technology could be developed to operate at higher temperatures and pressures. The primary incentive is to improve the efficiency or increase the shaft power output of advanced technology steam turbines. DeLaval Turbine and Compressor Division personnel believe that technology consistent with inlet steam conditions of 1800 psig and 1050°F can be developed and brought to commercial service in the 1985 - 2000 period. Improvements in materials and casing construction contribute to these developments.

The study encompasses four steam turbine - heat source combinations. Two configurations using liquid fuel, either petroleum or coal-derived boiler fuel, incorporate current and advanced turbine technology. The other two also include a current and an advanced technology, but coal is used in both cases. For the current technology, a coal furnace is employed with flue gas desulfurization. The advanced system uses an atmospheric fluid bed coal combustion system.

PERFORMANCE CHARACTERISTICS

The performance of automatic extraction turbines is typically presented in terms of both the throttle and extraction steam flow as a function of shaft power output. Figure III-5 presents the performance of a current technology 18 megawatt turbine-generator with 600 psig extraction and 3 psia condensing pressures. The highest utilization of the energy supplied in the steam occurs with maximum extraction (minimum flow to the condenser). The maximum throttle flow occurs with maximum extraction, and the minimum occurs with zero extraction at a given shaft power output. If the shaft power were all produced by extracted steam, the low

pressure section of the turbine would idle and windage heating could be harmful. To prevent this, a small steam flow always passes through the low pressure turbine section to the condenser.

The steam turbine offers operating flexibility. The shaft power output can be varied over a wide range including 10 percent overload. At part power, the upper limit on steam flow is determined by the performance curve (constant extraction flow) and the minimum flow to the condenser in Figure III-5.

The extraction steam turbine provides design flexibility; almost any electrical-thermal requirement can be satisfied. Figure III-6 illustrates this characteristic for the advanced technology turbine with 600 psig extraction pressure and 3 psia condensing pressure at 18 megawatts generator power output. If the industrial process requires large amounts of steam in relation to the electrical need, high extraction is appropriate. Processes with high power requirements in comparison to the thermal needs would operate with lower extraction flows. Of necessity, emphasis on electrical power output reduces the overall utilization of the fuel provided.

In order to provide a wide range of possibilities to match each industrial application, a total of 10 steam turbine design options were selected for each technology-thermal source combination. Half of the options extracted steam at 600 psig and half at 50 psig. The performance of the steam sources and the parasitic thermal and electric loads were included in the calculations. The performance of each design option was evaluated over a range of sizes based upon detail design and performance data provided by DeLaval for the turbine generator at 6 and 18 megawatts and provided by Bechtel National, Incorporated for heat sources and balance-of-plant (Volume IV). The design point was 18 megawatts; the minimum size, 0.6 megawatts; and the largest was listed as 30 megawatts. Actually, the steam turbine can be considerably larger, but 30 megawatts did not limit the results.

The design point energy conversion system performance data for each design option, turbine technology and fuel combination are tabulated in Table VI-10, Volume VI, pages 411-414 and 419-422 - a total of 40 steam turbine design options. Figure III-7, which is representative of a set of design options, indicates the electric output and extraction steam energy in terms of the higher heating value of the fuel consumed. The variation in electrical-to-thermal ratio varies from 2.4 to 0.25 for this set of design options. Table III-2 indicates design point electrical-to-thermal ratios and other information for both current and advanced technology design options. The data presented in Figure III-7 corresponds to the information in the last column in Table III-2.

At each design point, the turbine-generator produces 18 megawatts of electrical power. Based on corresponding data at 6 megawatts, the fuel utilization for both steam and electricity have been defined from 0.6 to 30 megawatts. Above 30 megawatts the turbine generator performance characteristics were assumed not to vary with size. Figure III-8 is an example of this variation for design option 5 of the advanced technology system with the coal heat source. Another statement of the size effects is presented in Figure III-9 which indicates the efficiency variation with size at zero extraction (Maximum electrical output).

ESTIMATED COST

The turbine-generator costs were provided by DeLaval based upon 1978 prices for similar equipment conforming to 1978 NEMA standards. The costs include the turbine, generator, gear box (required for small units), lubricating system, controls and basic accessories. The costs of the current technology single automatic extraction steam turbine-generator are presented in Figure III-10 as a function of size. The cost data for the advanced technology is presented in Figure III-11. The installation cost for the turbine-generator is included as balance-of-plant System 14 in Volume IV as a function of the equivalent horsepower rating of the equipment. Figure III-12 presents this Bechtel National estimate as a function of electrical output in megawatts.

The condenser cost estimates are based on published data and private information. The condenser costs employed in this study, Figure III-13 are stated in terms of the quantity of steam to be condensed. Those data were converted to costs per kilowatt output from the turbine-generator in Figure III-14. The installation costs shown in Figure III-14 are based upon the balance-of-plant equipment installation costs in Volume IV for erected equipment (tanks, vessels, etc.).

The complete steam turbine energy conversion system cost includes the heat source, balance-of-plant items, contingencies and fees. The magnitude of these various cost elements is illustrated in Figure III-15 where the installed capital cost of the representative 18 megawatt steam turbine system is broken down. The turbine-generator and condenser typically represent about 20 percent of the complete energy conversion system cost. The variation of the cost of this representative system with size is included in Figure III-16.

The operating and maintenance cost (excluding fuel) was estimated to be 0.06 ¢/kWh for the turbine, generator and condenser.

EMISSIONS

The emissions from the heat sources and the waste materials involved in coal-fired steam turbine systems are defined in Volume IV. Table III-3 presents those emissions in terms of the higher heating value of the fuel consumed.

PHYSICAL CHARACTERISTICS

The turbine and generator weights are presented in Figure III-17. The floor area requirements of the steam turbine-generator are included in Figure III-18. The condenser typically requires 2 square feet for each 1000 pounds of steam condensed per hour.

COGENERATION APPLICABILITY

Steam turbines have wide cogeneration applicability as is evident by their use in industry today. Since the turbine can operate with steam generated from a variety of heat sources using a variety of fuels, it offers fuel flexibility and can use coal, coal derived fuel, alternate fuels, or process by-product fuels. The extraction type turbine is able to meet a wide variety of industrial process requirements.

The Cogeneration Technology Alternatives Study reported here has used a thermal bin system to serve as a framework to define industrial requirements and also energy conversion system characteristics. Steam temperatures of 300, 500 and 700°F are common in industrial plants and served for this study. However, 300°F corresponds to a steam pressure of 50 psig while 500°F has a saturation pressure a little above 600 psig. Process requirements between 300 and 500°F have been met with 600 psig steam. Since extraction pressure has a dominant influence on the electrical energy available, processes with lower temperature steam requirements, for example 370°F (190 psig) in the paper industry, could be better served with lower pressure steam than provided by the bin system (600 psig).

The analysis procedure in evaluating an energy conversion system in an industrial cogeneration application first determined the fuel energy savings for each design option and selected the most conserving for further analysis. In many cases requiring steam above 50 psig, the most energy conserving configuration extracted steam from the turbine at 50 psig to preheat water fed to a separate conventional boiler which provided the process steam. The alternate design with 600 psig extraction pressure did not provide efficient use of the higher pressure steam when, for example, 130 or 200 psig was required. In effect, the bin system did not provide a fine gradation which would best use the flexibility of the extraction steam turbine. The bin system was used in the computer analysis and the results presented in Volume VI. However, these results were modified to account for the effects of extracting steam at a pressure matching the process requirements. The modified data for those industrial processes affected are presented in Tables III-4

through III-7 for the match electric strategy. These results have been incorporated in the program summary presented in Volume I.

The match thermal requirement may appear to be fairly straightforward, but actually this choice is complex with the infinite bin system. A turbine performance curve, such as Figure III-6, can be drawn for any extraction pressure appropriate to the process. Figure III-19 presents the advanced steam turbine performance in terms of the higher heating value of the fuel for various extraction pressures. The data in Figure III-19 are based on the performance of a coal-derived liquid boiler fuel heat source and appropriate parasitic losses. Maximum fuel utilization occurs at maximum extraction, but the corresponding process thermal and electrical energy requirements may not match the extracted steam energy and turbine-generator electrical output. If the thermal energy requirements are met and the electrical output is not sufficient for the process, the steam flow through the turbine to the condenser could be increased with an increase in electrical output, or the needed electrical energy could be purchased from the utility. In general, the maximum extraction condition was selected in the computer calculations to provide the highest fuel utilization although the resulting turbine-generator output is often significantly smaller than the industrial process electrical energy requirement. The additional energy is purchased from the electric utility. In the match thermal cases reported in Volume I, the results of the computer analysis were modified on the basis of an infinite bin and operation at maximum extraction to provide the highest possible fuel savings. Tables III-8 through III-11 present the modified steam turbine data for the match thermal designs.

REFERENCES

1. B.G.A. Skrotski, "Steam Turbines," Power, June 1962.

2. "Richard Rapid System, Process Plant Construction Estimating Standards," 1978 - 1979 Edition.

3. M. W. Larinoff, "Look at Costs of Wet/Dry Towers," Power, April 1978.

TABLE III-2

STEAM TURBINE ENERGY CONVERSION SYSTEM
DESIGN POINT ELECTRICAL/THERMAL RATIOS

Technology	Current				Advanced			
Heat Source	Oil	Oil	Coal FGD	Coal FGD	Oil	Oil	Coal AFB	Coal AFB
Output-MW	18	18	18	18	18	18	18	18
Speed-RPM	3600	3600	3600	3600	3600	3600	3600	3600
Inlet Pressure, psi	1200	1200	1200	1200	1800	1800	1800	1800
Inlet Temperature, °F	950	950	950	950	1050	1050	1050	1050
Condenser Pressure, psia	3	3	3	3	3	3	3	3
Extraction pressure, psig	600	50	600	50	600	50	600	50
Design Option								
1 Electrical/thermal	0.52	2.15	0.52	2.06	0.83	2.38	0.85	2.44
2	0.20	0.85	0.20	0.85	0.33	0.98	0.33	0.98
3	0.10	0.42	0.09	0.41	0.16	0.48	0.16	0.49
4	0.06	0.28	0.06	0.28	0.10	0.32	0.10	0.32
5	0.05	0.22	0.04	0.22	0.08	0.26	0.08	0.25

TABLE III-3

STEAM TURBINE ENERGY CONVERSION SYSTEM
EMISSIONS AND WASTES
Pounds/Million BTU Fuel Input

Technology	Current		Advanced	
Fuel	Petroleum Boiler Fuel	Coal (FGD)	Coal-Derived Boiler Fuel	Coal (AFB)
SO_2	0.76	1.20	0.824	1.20
NO_x	0.50	0.70	0.50	0.20
Hydrocarbons	0.020	0.014	0.020	----
Particulates	0.016	0 100	0.100	0.100
Solid Wastes	---	0.76	0.053	36.0
Waste Water (Blowdown)	7.1	6.8	6.9	6.9

TABLE III-4

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH E
CURRENT STEAM TURBINE 950°F 1200 psi (ECS #1)
PETROLEUM BOILER FUEL

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	-0.285	-30.7	-0.607	-240.6
5	Saw Mill	-0.129	-14.6	-0.716	-366.3
6	Newsprint	-0.061	- 7.3	-0.171	- 74.5
7	Writing Paper	0.053	5.9	-0.047	- 22.9
8	Corrugated Paper	0.219	110.7	0.158	295.3
9	Boxboard	0.267	41.4	0.195	111.7
10	Chlorine/Caustic	0.083	-55.6	-0.207	-511.4
11	Alumina	-0.008	- 0.9	-0.067	- 29.5
15	Rubber	-0.013	- 0.2	-0.110	- 7.9
16	Nylon	-0.330	- 9.6	-0.529	- 56.4
20	Tires	-0.141	-15.6	-0.306	-123.7
25	Copper	-0.146	-23.4	-0.302	-177.9
26	Motor Vehicles	-0.079	-16.5	-0.257	-194.8

TABLE III-5

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH E
CURRENT STEAM TURBINE 950°F 1200 psi (ECS #2)
COAL FUEL

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	-0.272	-29.3	-0.466	-185.0
5	Saw Mill	-0.187	-21.0	-1.547	-791.8
6	Newsprint	-0.042	- 5.0	0.134	59.1
7	Writing Paper	0.099	32.2	0.277	136.0
8	Corrugated Paper	0.290	147.1	0.390	731.2
9	Boxboard	0.307	47.5	0.373	214.1
10	Chlorine	-0.071	-47.6	0.106	264.6
11	Alumina	0.010	1.1	0.088	44.1
15	Rubber	0.011	0.2	0.061	4.3
16	Nylon	-0.315	- 9.1	-0.233	- 24.9
20	Tires	-0.118	-12.8	-0.072	- 29.5
25	Copper	-0.121	-19.4	-0.034	- 20.0
26	Motor Vehicles	-0.057	-11.9	-0.067	- 50.5

TABLE III-6

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH E
ADVANCED STEAM TURBINE-1050°F 1800 psi (ECS #7)
COAL DERIVED BOILER FUEL

No.	Industry	FESR	Total Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile	-0.231	-24.9	-0.601	-238.2
5	Saw Mill	-0.222	-24.9	-0.965	-493.6
6	Newsprint Mill	+0.040	+ 4.8	-0.068	- 29.6
7	Writing Paper Mill	0.185	20.7	+0.067	+ 36.5
8	Corrugated Paper	0.399	201.6	0.331	622.1
9	Box Board	0.424	65.7	0.345	198.9
10	Chlorine	+0.002	+ 1.3	-0.122	-300.1
11	Alumina	+0.044	+ 4.9	-0.023	- 10.3
15	Butadiene Rubber	0.036	0.7	-0.081	- 5.8
16	Nylon	-0.273	- 7.9	0.491	- 52.4
20	Tires	-0.070	- 7.6	-0.243	- 98.7
25	Copper	-0.081	-12.9	-0.249	-146.4
26	Motor Vehicles	-0.020	- 4.2	-0.201	-152.6

TABLE III-7

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH E
ADVANCED STEAM TURBINE-1050°F 1800 psi (ECS #8)
COAL (AFB)

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	-0.218	-23.5	-0.407	-181.4
5	Saw Mill	-0.284	-31.8	-1.238	-633.1
6	Newsprint	0.057	6.9	0.245	106.7
7	Writing Paper	0.220	24.5	0.324	158.8
8	Corrugated Paper	0.430	217.7	0.521	977.4
9	Boxboard	0.455	70.5	0.514	295.6
10	Chlorine	0.014	9.3	0.209	517.4
11	Alumina	0.061	6.8	0.143	63.7
15	Rubber	0.058	1.1	0.132	9.5
16	Nylon	-0.258	- 7.4	-0.158	- 16.9
20	Tires	-0.047	- 5.2	0.056	23.2
25	Copper	-0.062	-10.0	-0.67	39.0
26	Motor Vehicles	-0.003	- 0.6	0.059	44.4

TABLE III-8

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH T
CURRENT STEAM TURBINE 950°F 1200 psi (ECS #1)
PETROLEUM BOILER FUEL

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	0.006	0.7	-0.029	-11.4
5	Saw Mill	—	—	—	—
6	Newsprint	0.036	4.3	0.012	5.2
7	Writing Paper	0.065	7.3	0.021	10.3
8	Corrugated Paper	0.317	160.5	0.255	477.9
9	Boxboard	0.317	49.0	0.258	147.5
10	Chlorine/Caustic	0.018	12.2	-0.002	-4.3
11	Alumina	0.039	4.3	0.015	6.6
15	Rubber	-0.008	-0.2	-0.080	-5.7
16	Nylon	-0.036	-1.1	-0.074	-7.8
20	Tires	-0.026	-2.8	-0.065	-26.5
25	Copper	0.012	1.9	-0.033	-19.3
26	Motor Vehicles	0.008	1.5	-0.067	-51.0

TABLE III-9

STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH T
CURRENT STEAM TURBINE 950°F 1200 psi (ECS #2)
COAL FUEL

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	0.034	3.9	0.029	9.6
5	Saw Mill	—	—	—	—
6	Newsprint	0.087	10.5	0.109	48.0
7	Writing Paper	0.176	19.7	0.190	92.8
8	Corrugated Paper	0.454	229.9	0.466	876.1
9	Boxboard	0.477	73.8	0.443	255.1
10	Chlorine	0.048	32.3	0.088	217.8
11	Alumina	0.035	3.9	-0.025	-11.1
15	Rubber	0.065	1.4	0.097	7.0
16	Nylon	-0.012	-0.4	0.016	1.8
20	Tires	0.020	2.2	0.056	22.8
25	Copper	0.056	8.9	0.079	52.9
26	Motor Vehicles	0.061	12.4	0.034	26.6

TABLE III-10

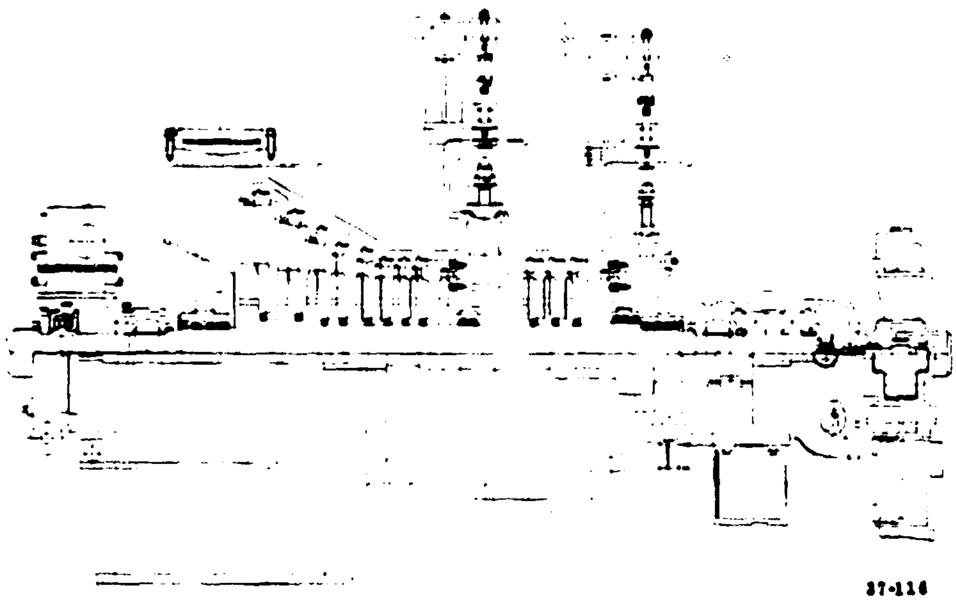
STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH T
ADVANCED STEAM TURBINE-1050°F 1800 psi (ECS #7)
COAL DERIVED BOILER FUEL

No.	Industry	FESR	Total Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile	0.019	2.1	-0.018	-7.2
5	Saw Mill	—	—	—	—
6	Newsprint Mill	0.091	10.7	0.055	24.0
7	Writing Paper Mill	0.140	15.6	0.080	39.0
8	Corrugated Paper	0.491	255.1	0.421	787.9
9	Box Board	0.449	69.4	0.379	218.1
10	Chlorine	0.033	22.3	0.012	30.8
11	Alumina	0.003	0.4	-0.019	-8.5
15	Butadiene Rubber	0.028	0.6	-0.059	-4.3
16	Nylon	0.029	0.8	-0.016	-1.7
20	Tires	0.007	0.8	-0.041	-16.9
25	Copper	0.033	5.3	-0.034	-19.9
26	Motor Vehicles	0.037	7.9	-0.059	-44.5

TABLE III-11

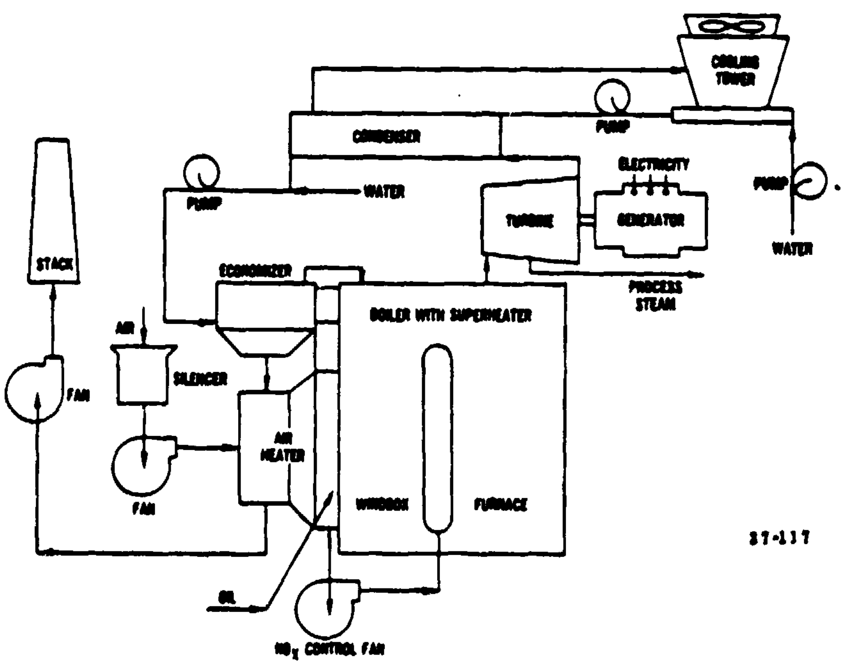
STEAM TURBINE DATA FOR PROCESS PRESSURE
PROCESS LEVEL-MATCH T
ADVANCED STEAM TURBINE-1050°F 1800 psi (ECS #8)
COAL (AFB)

No.	Industry	FESR	Process Fuel Savings 10^{12} BTU	Cost Savings Ratio	Cost Savings Million Dollars
4	Textile Mill	0.022	2.2	0.051	20.3
5	Saw Mill	—	—	—	—
6	Newsprint	0.059	7.1	0.117	51.1
7	Writing Paper	0.105	11.8	0.210	102.6
8	Corrugated Paper	0.356	196.0	0.466	874.7
9	Boxboard	0.378	58.6	0.457	262.8
10	Chlorine	0.034	22.5	0.098	244.5
11	Alumina	-0.006	-0.7	0.106	46.6
15	Rubber	0.030	0.6	0.127	9.2
16	Nylon	-0.025	-0.7	0.035	3.7
20	Tires	-0.005	-0.5	0.120	48.5
25	Copper	0.034	5.7	0.146	85.8
26	Motor Vehicles	0.033	6.8	0.086	65.6



37-116

Figure III-1. DeLaval Extraction Turbine



37-117

Figure III-2. Steam Turbine Conversion System - Petroleum Boiler Fuel

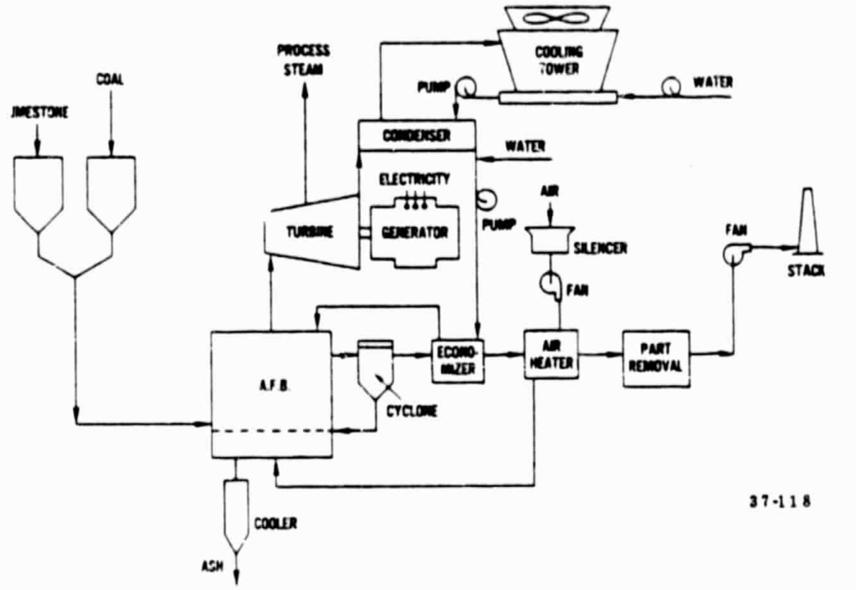


Figure III-3. Steam Turbine Conversion System - Coal Fired Atmospheric Fluid Bed Boiler

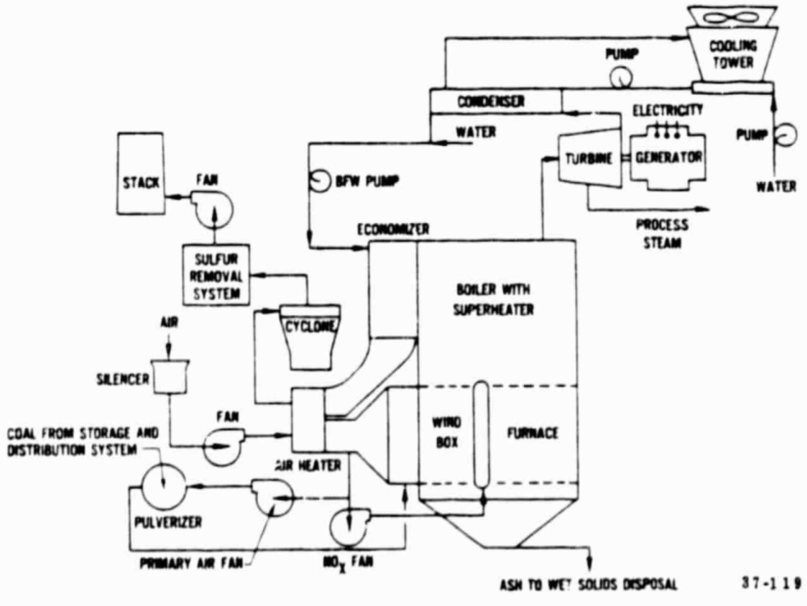


Figure III-4. Steam Turbine Conversion System - Coal Fired Boiler With Exhaust Gas Clean-up

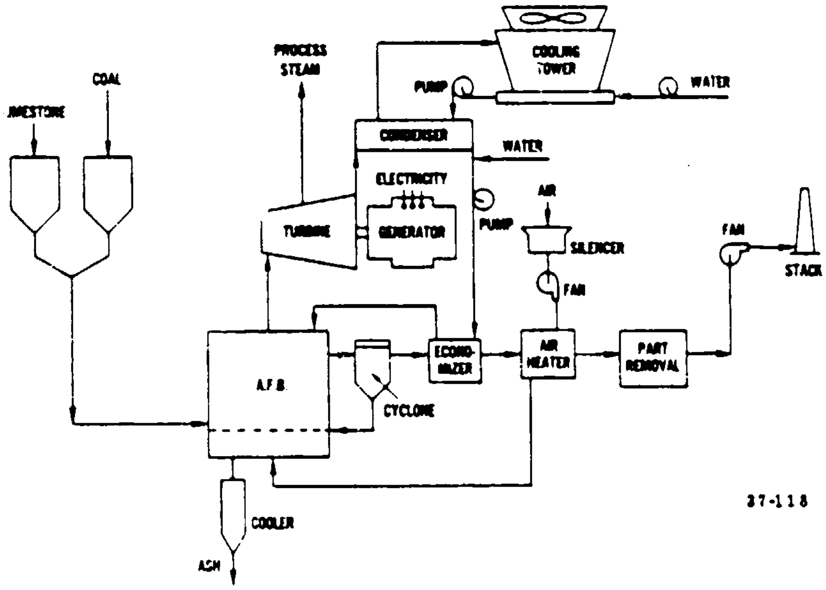


Figure III-3. Steam Turbine Conversion System - Coal Fired Atmospheric Fluid Bed Boiler

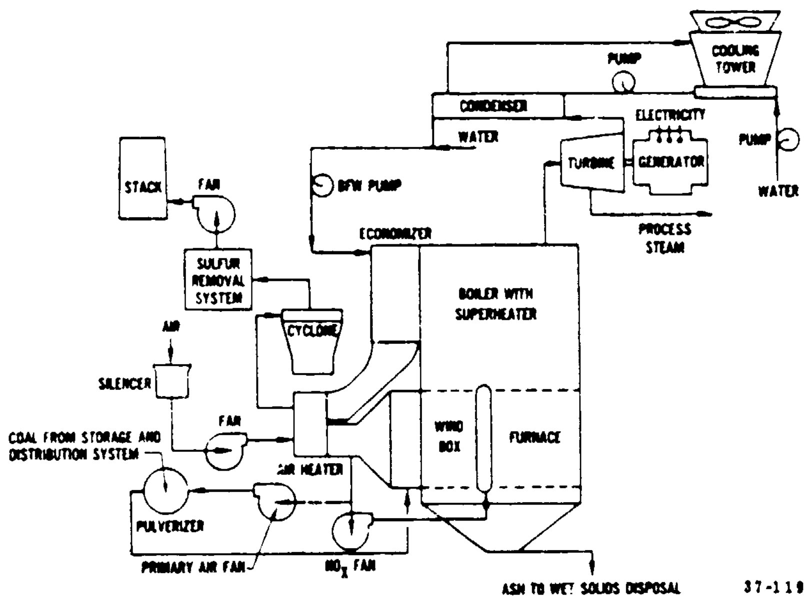


Figure III-4. Steam Turbine Conversion System - Coal Fired Boiler With Exhaust Gas Clean-up

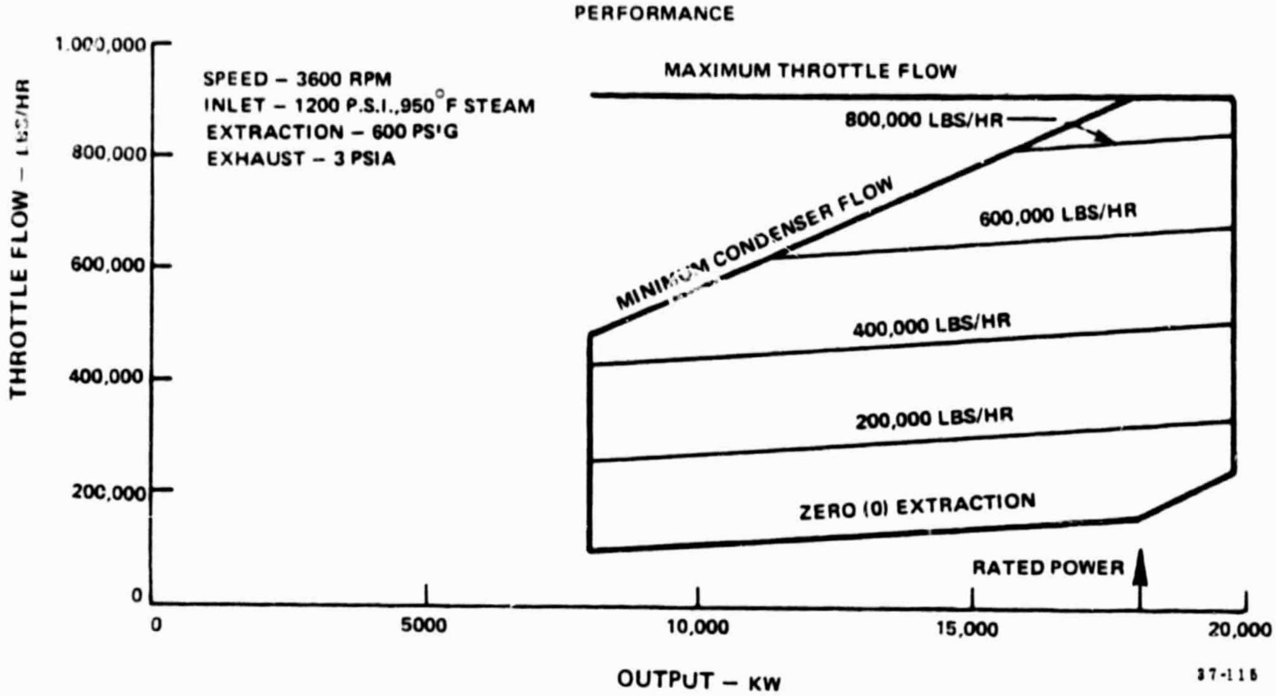


Figure III-5. Single Automatic Extraction Condensing Turbine-Generator

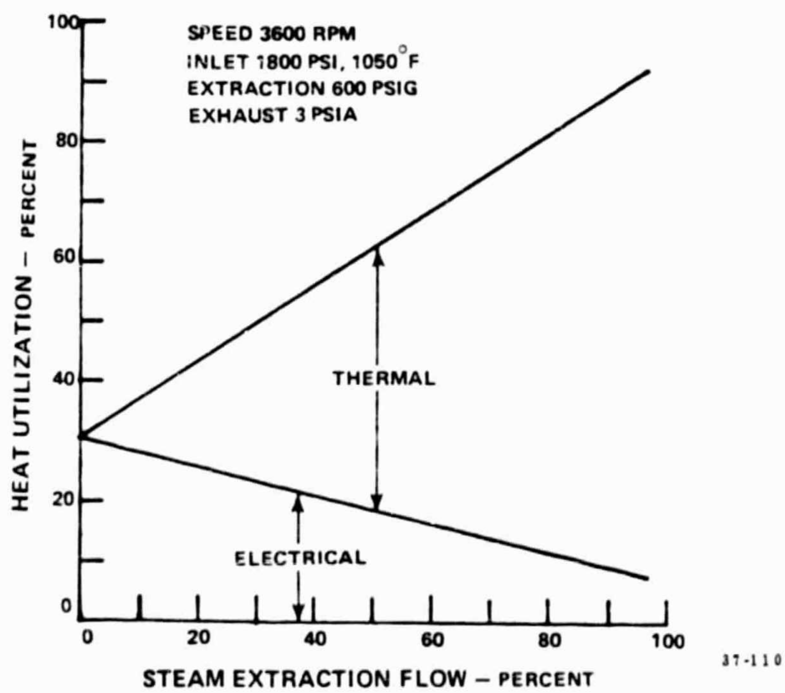


Figure III-6. Steam Turbine and Generator Performance

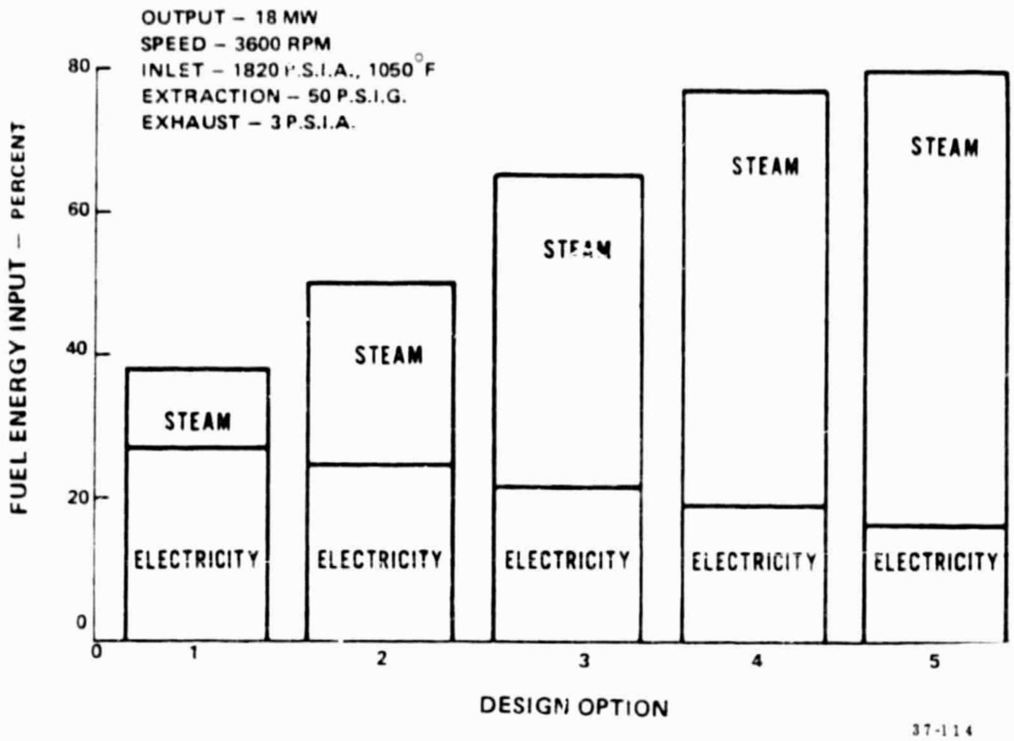


Figure III-7. Steam Turbine Performance - Coal Fired Atmospheric Fluidized Bed

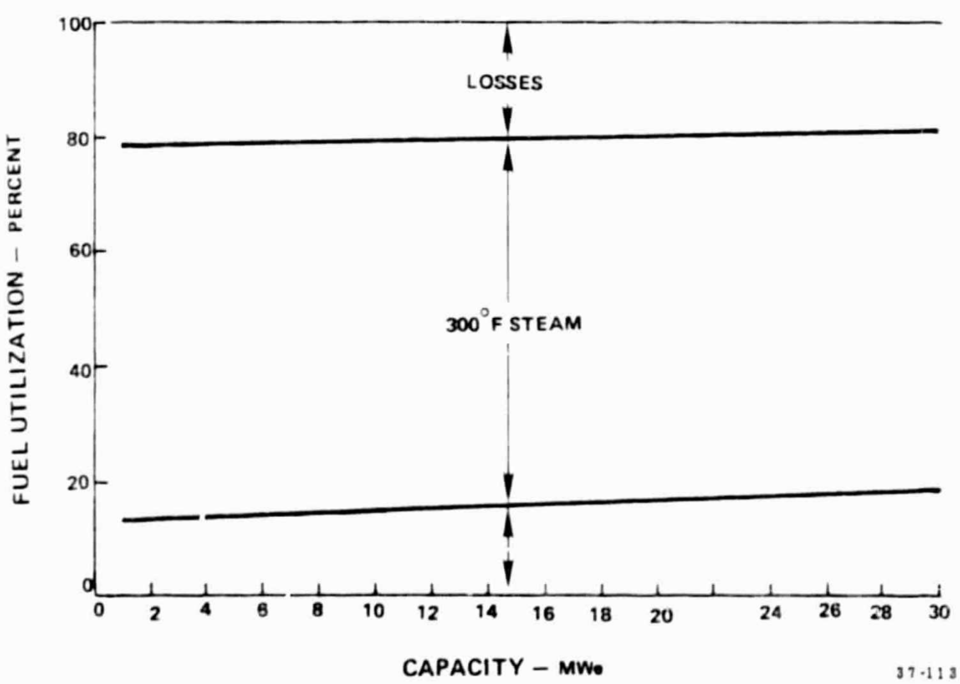


Figure III-8. Steam Turbine Performance - Coal Fired Atmospheric Fluidized Bed - Design Option No. 5

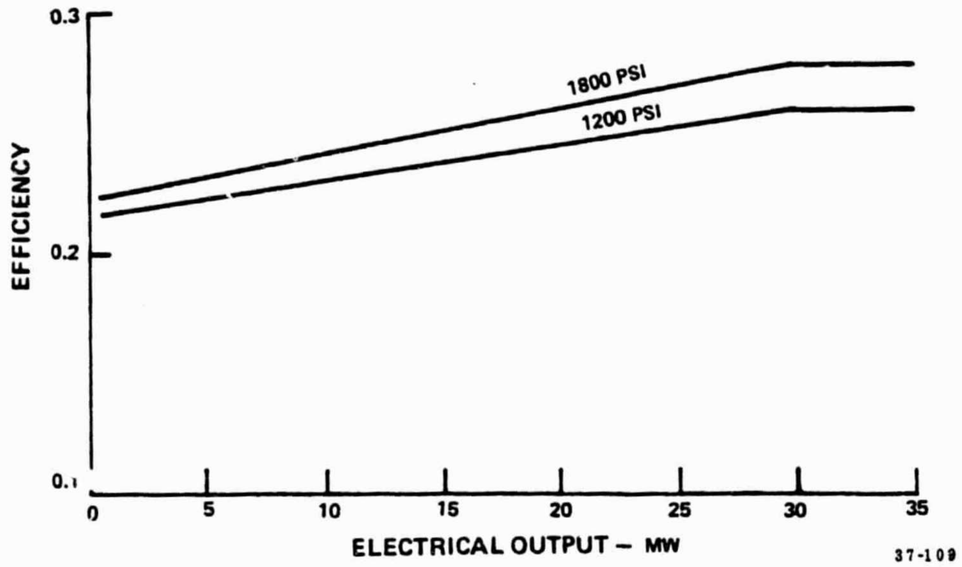


Figure III-9. Steam Turbine Performance - Zero Extraction-Coal-Derived Boiler Fuel

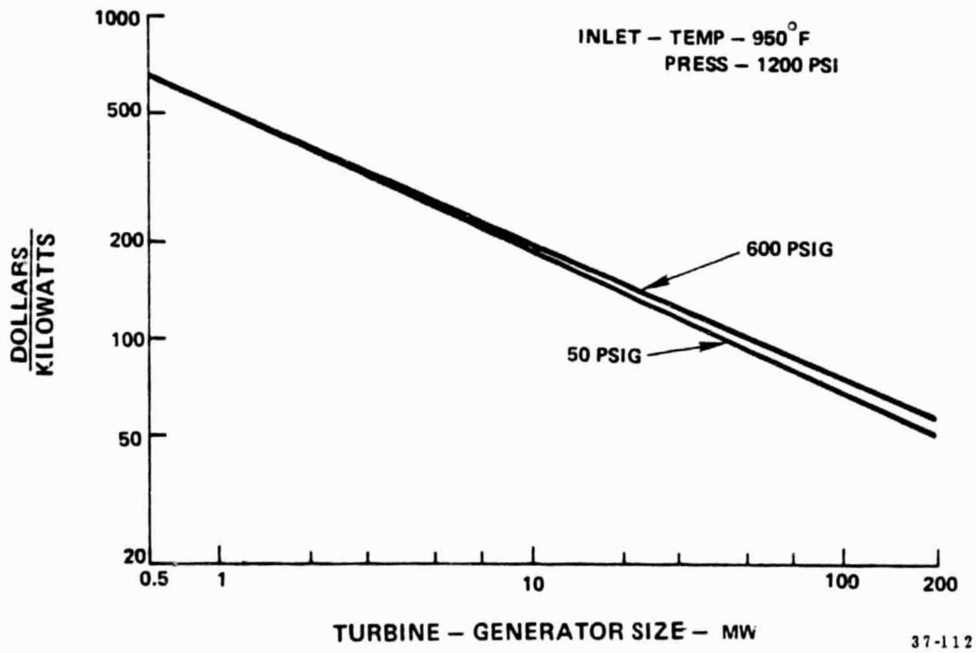


Figure III-10. Steam Turbine - Generator Equipment Costs

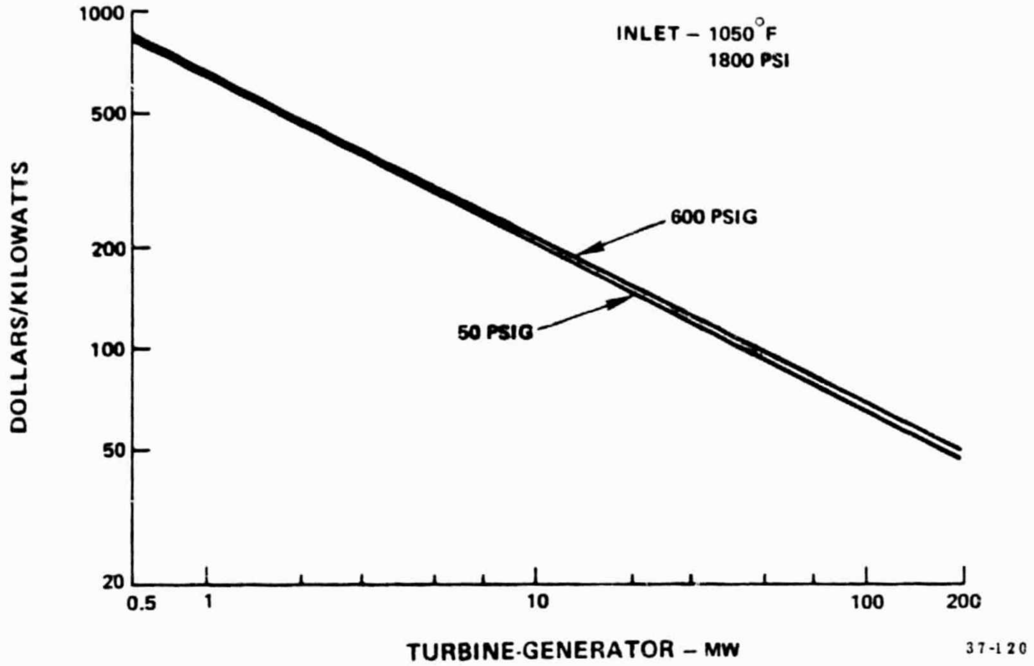


Figure III-11. Steam Turbine - Generator Equipment Costs

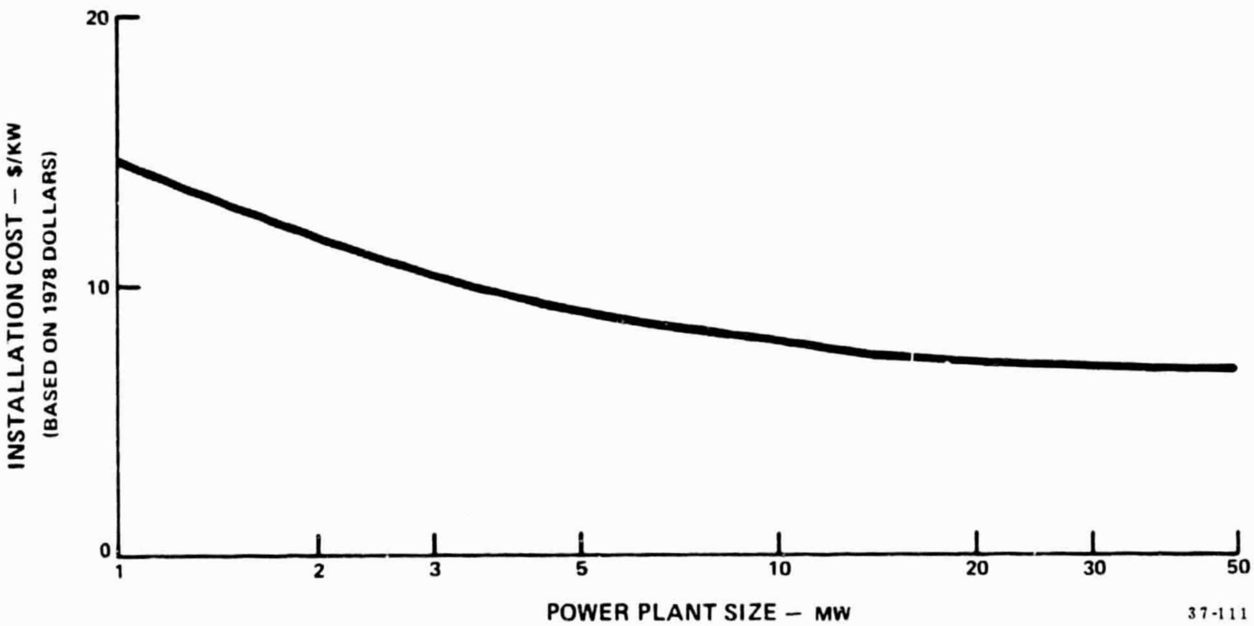


Figure III-12. Steam Turbine - Generator Installation Cost

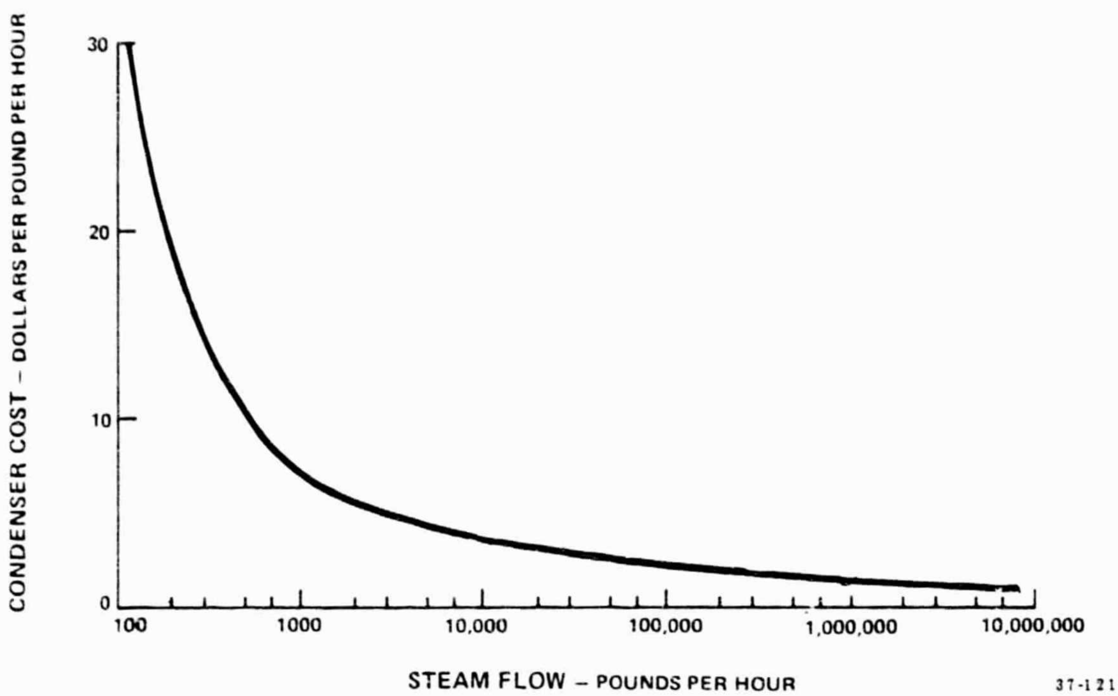


Figure III-13. Steam Condenser Costs - 1978 Dollars

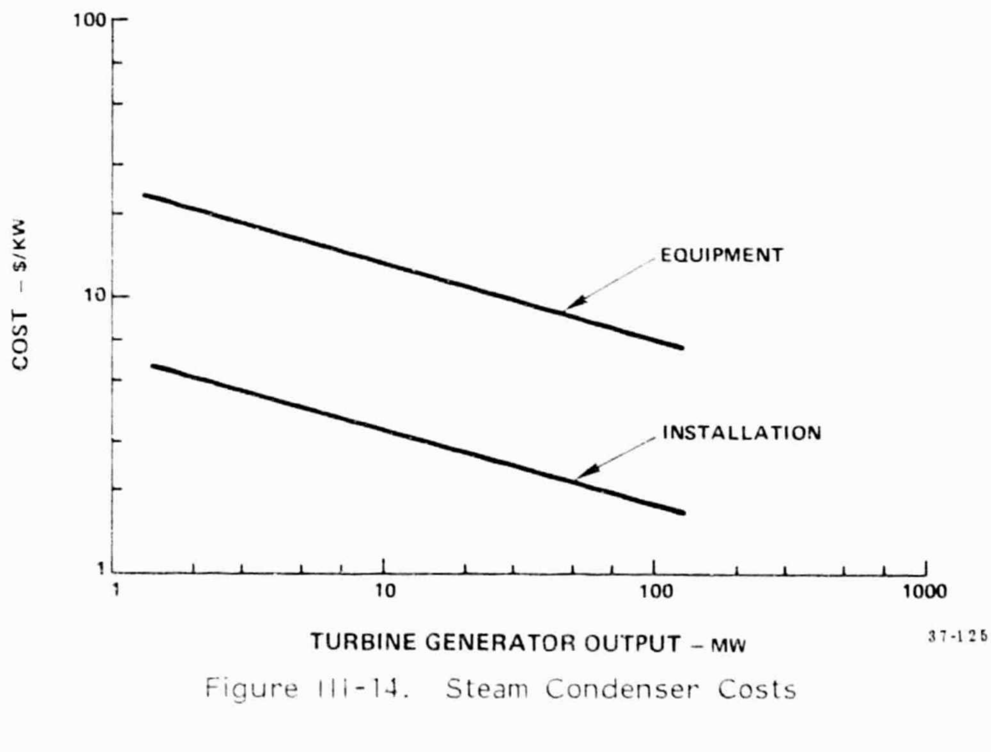


Figure III-14. Steam Condenser Costs

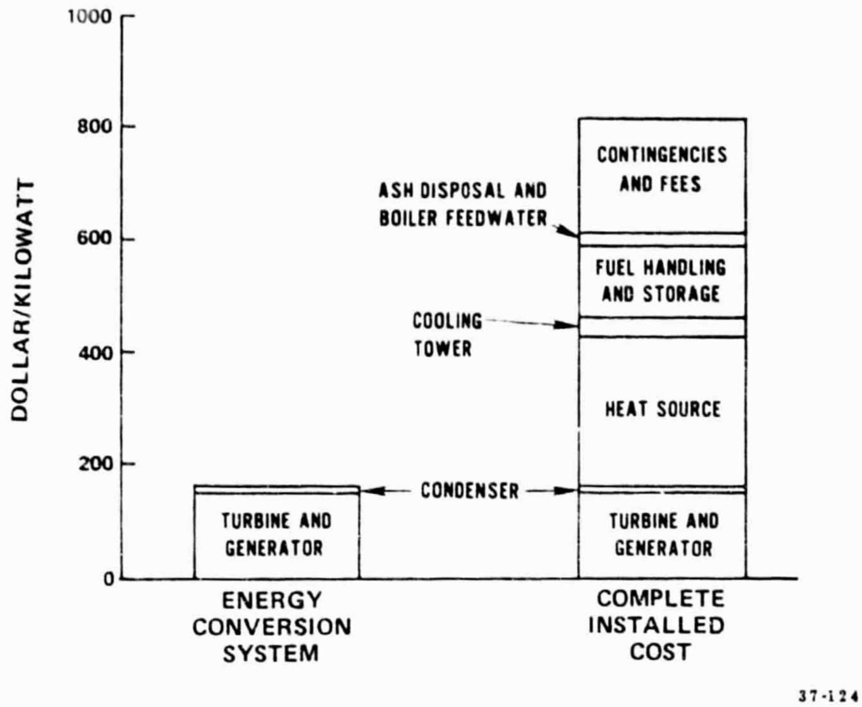


Figure III-15. Steam Turbine System Capital Cost - Coal Fired Atmospheric Fluidized Bed

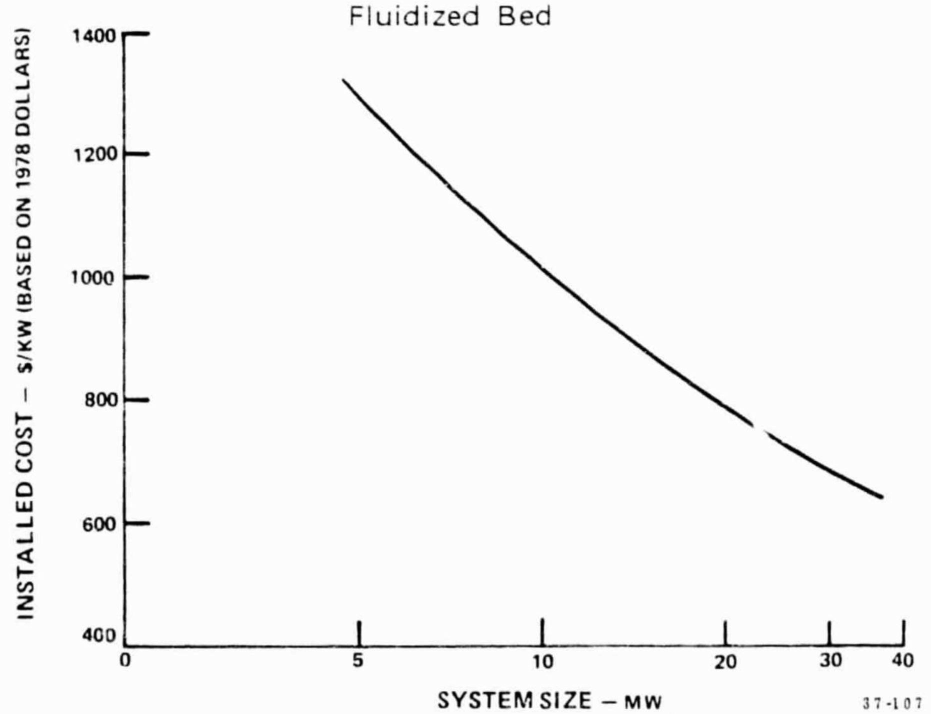


Figure III-16. Steam Turbine System Cost Variation - Coal Fired Atmospheric Fluidized Bed

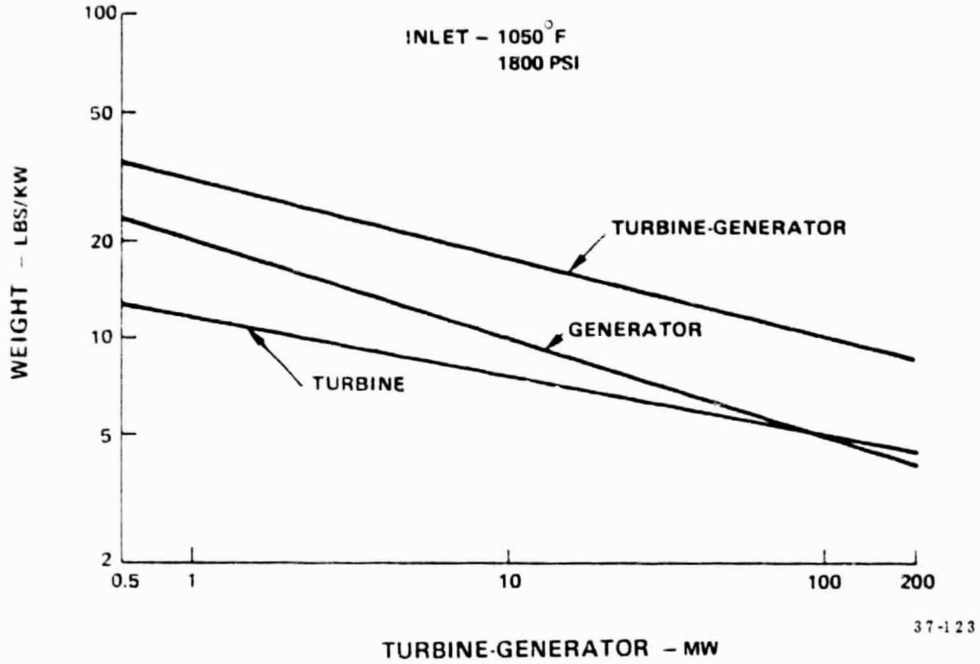


Figure III-17. Steam Turbine - Generator Weights

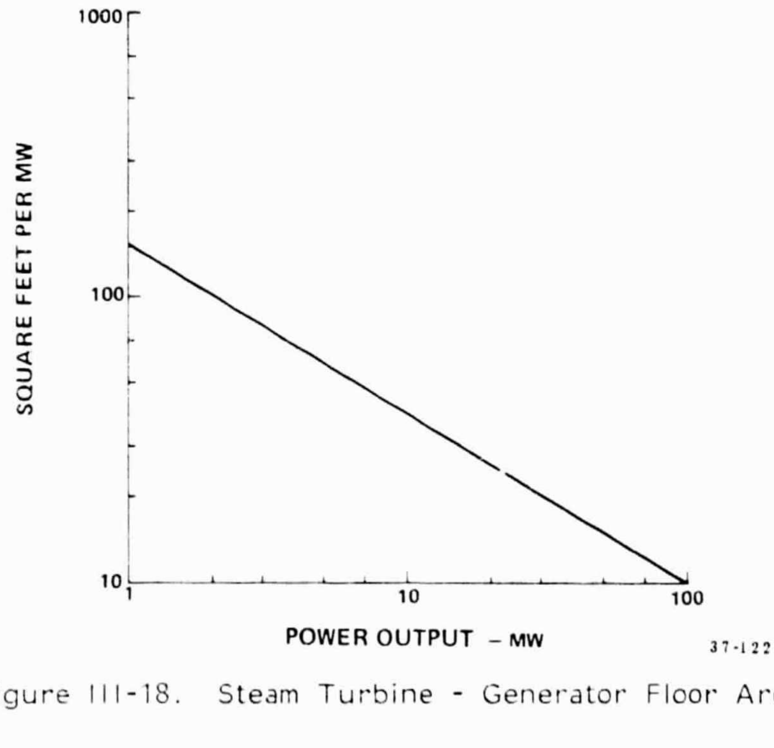


Figure III-18. Steam Turbine - Generator Floor Area

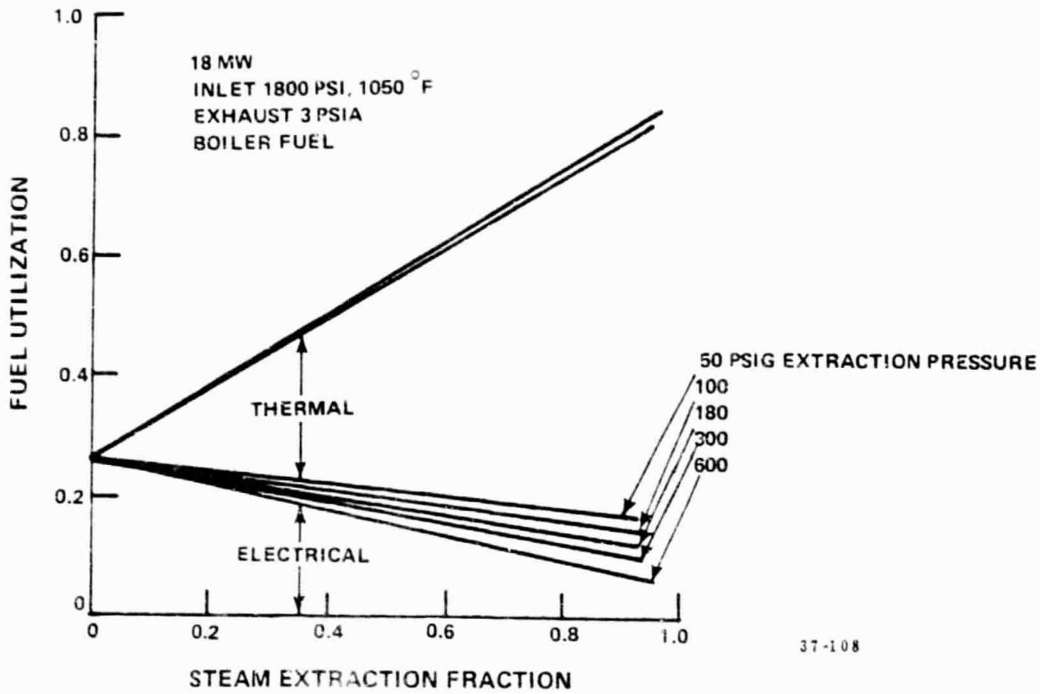


Figure III-19. Steam Turbine Performance

DIESEL GENERATORS

INTRODUCTION

Diesel generators utilize an internal combustion engine as the prime mover coupled to a 3-phase alternator to produce electric power. They can be used for cogeneration by recovering heat from the prime mover cooling system and exhaust gases.

Diesel engine technology can be broadly classified by three types: high, medium and low speed. The high speed diesel engines in this study are four cycle machines that generally operate between 1200 and 3600 rpm and produce up to 1½ MW of electrical power. Most of the engines in this classification are used for vehicle propulsion including commercial vehicles such as trucks, passenger cars, small marinecraft, agricultural equipment and rail locomotive applications. They are also used for electrical generation and some cogeneration with total energy applications and for auxiliary power sources.

Medium speed diesel engines are also four cycle devices used for marine propulsion, rail locomotives, rural electrification and stationary emergency power. They generally operate between 500 and 1200 rpm and produce outputs in the range of 1/2 to 10 or more megawatts. The high speed and medium speed diesel engines considered in this study typically operate at about 200 psi brake mean effective pressure.

The large, low speed diesel engines used in this study have historically been used for marine propulsion applications. They have also been used for stationary electric power plants. The low speed engines typically are two cycle devices operating at approximately 120 to 180 rpm. Typical sizes range from under 10 MW to 30 MW. These large machines use up to 3 foot diamater pistons operating through a stroke of up to 5 feet. They typically operate at 180 psi brake mean effective pressure.

Current research efforts directed towards high temperature diesel engine operation, which could be most attractive for cogeneration application, appear to be most advanced for high speed engines. The low speed machines are most advanced in using heavy oils and coal fuels. Therefore, both of these types of diesel engine-generators are included in the study.

LOW SPEED DIESEL ENGINE GENERATORS

Conversion System Description

The current and advanced low speed diesel engine design characteristics used in this study are based on data supplied by Sulzer Brothers, Limited, of Switzerland supported by Dr. P. S. Myers of the University of Wisconsin. The assumption was made that the latest engine introduced by Sulzer Brothers represents technology which will be available in the 1985-2000 period. This assumption is based on prior Sulzer Brothers experience presented in Figure III-20 that indicates that 10 to 15 years are necessary for a new engine to reach full commercial use. Therefore, the engine which entered the market place in the mid 1970's is representative of commercially available low speed diesel technology in the late 1980's. The advanced technology engine incorporates improvements to accommodate coal derived boiler fuel and powdered coal and to provide higher cooling water temperatures for cogeneration.

The low speed diesel prime mover is a multi-cylinder, single acting, two stroke engine with cross head and exhaust gas turbo charging operating at 120 rpm. A cross sectional drawing of such an engine is shown in Figure III-21 and the engine design parameters are presented in Table III-12.

The engine is water cooled and typically about 1750 BTU per brake horsepower-hour are removed by this sytem. A breakdown of the cooling requirements is presented in Table III-13. A closed system is provided for cylinder and cylinder cover cooling. Oil cooling of the piston is no longer adequate for the high mean

effective pressures of modern supercharged, two stroke engines. A closed separate cooling water circuit for piston cooling is used. Water cooling with its better heat transfer coefficient provides much lower piston temperatures. Any lubricating oil penetrating is removed by an efficient oil separator. A heat exchanger for lubricating oil cooling is included in the water system. The combustion air is cooled before it enters the working cyclinders. The charge air coolers in the air receiver are connected to the water system.

Sulzer Brothers have designed a 22 MW electric generating station incorporating a 12 cylinder low speed diesel engine. This design intended for electric utility application serves as the basis for the cogeneration diesel engine design in this study. The external view of the electric generating station is shown in Figure III-22. Also shown by the dotted lines are provisions for later expansions by the addition of another diesel engine generator.

A more detailed drawing of the 22 MW electric generating station is shown in Figure III-23. The engine generator set with its auxiliary equipment and water cooling plant constitutes a completely independent unit. The diesel engine (1) and the three phase alternator (2) are directly coupled and operate at the same speed.

The exhaust gases from the diesel engine are directed into a silencer (3) which is combined with the exhaust gas boiler (5) an additional silencer (4) reduces the exhaust noise to acceptable level, if required. Cooling towers (6) are provided as necessary.

The diesel engine is started by compressed air stored in reservoirs (7) at 30 atmospheres.

The fuel treatment plant which is required when the engine runs on boiler type fuel consists of centrifuges (10) and service and mixing tanks (9).

Water coolers and oil coolers (11) are mounted as one unit. Water is supplied to these coolers by centrifugal pumps (13). The cooler, the pumps, and the tank for the separate fuel valve cooling circuit are mounted as a separate unit (14). The

piston cooling group (15) comprises pumps, coolers, and oil separators. The bearing lubricating oil is collected in a drain tank (16) and recirculated to the bearings via the oil cooler (11). Oil storage (17) consists of the tank for the cylinder lubricating oil and the feed pump for the automatic cylinder lubrication. The bearing lubricating oil system of the turbocharger is connected to the bearing lubricating oil system of the engine. An elevated tank (18) ensures the lubrication of the exhaust gas turbocharger when running down after stopping the engine.

The boiler type fuels must be preheated. The steam required for this purpose is raised in an exhaust gas boiler (5) supplemented by an oil fired boiler (19) when running at low outputs or for raising steam when the engine is started. A small emergency generating set (20) ensures the supply of all auxiliaries before the main diesel engine is started.

The cooler (21) for removing heat from the closed air circuit of the alternator cooling system is located below the alternator.

The auxiliary building includes the control desk, transformers, distribution panel for the auxiliaries, and instrumentation panels.

A diesel power station building accommodating large generating units can be expensive if the structural steel work of the machinery hall has to take the weight of an overhead traveling crane capable of lifting the heaviest part of the engine. On this basis, a 12 cylinder engine to provide 22 MW of electrical output would require a crane with a lifting capacity of 85 tons. Generally, a gantry crane of 85 tons capacity is provided during erection by the engine supplier. This mobile crane is designed to move heavy parts such as the crankshaft, base plate, rotor and stator from trucks into the machinery building. For this power plant, a 20 ton service crane installed in the building is adequate for subsequent maintenance work.

The combustion air for the diesel engine is taken straight from the building. Fresh air is constantly drawn through louvers. At maximum engine output about 15 air changes per hour take place. This ventilation system is self regulating and experience indicates that the room temperature can be maintained within acceptable limits.

Performance Characteristics

The current low speed diesel engine-generator can provide thermal energy by recovering heat from the 540°F exhaust gases and the cooling water system which operates at about 160°F. Figure III-24 presents a schematic of the current technology diesel engine and appropriate heat recovery equipment. The oil, piston, and cylinder cooling provide hot water. The charge air cooling provides both hot water and boiler preheat. The turbine exhaust produces low pressure steam in a heat recovery boiler. The symbols used in Figure III-24 are identified in Table III-14.

The cooling water temperature is raised to 265°F in the advanced technology low speed diesel engine. The heat recovery system, which is shown schematically in Figure III-25, provides 500°F steam as well as 300°F steam and hot water. (The symbols used in Figure III-25 are identified in Table III-14.) The oil and piston cooling provides hot water as in the current technology system. The cylinder cooling water at 265°F is used to preheat the high pressure and low pressure boilers. The charge air from the compressor is used to provide low pressure steam and preheat for the high pressure steam. The turbine exhaust contributes to both the high pressure and low pressure steam supplies.

The heat recovered from the various streams in the 12 cylinder engines of current and advanced technology are presented in Figure III-26. The turbine exhaust boiler, the charge air cooler, and the cylinder cooling water are the principle sources of heat. Figure III-27 presents the heat recovery in the thermal bins used in the study, hot water, 300°F steam, and 500°F steam. Figure III-28 presents similar data in terms of mass flow of water. The current technology does

not provide 500°F steam but produces more hot water than the advanced system. The advanced configuration provides 4 percent more heat overall and about 45 percent more steam. Figure III-29 presents the electrical output and total output of the advanced and current low speed diesel systems in terms of higher heating value of the fuel consumed. The advanced technology represents a small improvement in electrical efficiency and about 3 percent overall improvement in fuel utilization.

An alternate design for the advanced technology diesel-generator heat recovery system was included as design option number 2 of energy conversion system number 10 in Table III-1. In this case, the cylinder cooling water path was modified which resulted in reduced low pressure steam and more hot water.

The current technology low speed diesel-generator (energy conversion system number 4) and the advanced technology diesel-generator (energy conversion system number 10) use petroleum or coal derived boiler fuel. Low speed diesel-generators have been operated on an experimental basis on powdered coal. The assumption was made that such powdered coal systems could reach commercial application in the 1985-2000 period with performance similar to the diesel engine using boiler grade liquid fuel. The diesel system with coal fuel is designated energy conversion system number 11. There is a question concerning the ability to operate on coal with high sulfur content. However, for this study, the assumption was made that the specified high sulfur coal could be used after flotation type sulfur removal without performance or cost penalty. The design point performance used in the study is presented in Volume VI, Table VI-10.

A nominal design size of about 20 megawatts electrical output was used for the low speed diesel-generators. As indicated in Figures III-30 and III-31, there is virtually no performance variation for various size designs from 8 to 28 megawatts. The principal design variation over this range is the choice of number of cylinders in the engine from 4 to 12.

Low speed diesel generators provide high efficiency from 25 percent to rated electrical output, as indicated by the off-design performance data presented in Figure III-32. These data are representative of both current and advanced diesel systems. Typical design practice is to operate at 90 percent of rated power and the performance data used in this study are based on that practice.

Estimated Costs

The low speed diesel engine costs were provided by Sulzer Brothers for four sizes from 7 to 25 megawatts electric output based on operation of 90 percent of rated power. The cost estimates for the basic engine assume manufacture in Europe and delivery to an East Coast port. The appropriate customs duties are included. The generator costs are based upon manufacture in the United States.

Figure III-33 presents the costs of the diesel engine, generator, heat recovery equipment, and installation for the current technology (energy conversion system number 4). The corresponding data for the advanced systems (energy conversion systems numbers 9 and 10) are presented in Figure III-34.

The installation costs are based upon detailed estimates of labor requirements by Sulzer Brothers and the more general correlations provided by Bechtel National in Volume IV. A schedule of installation activities is presented in Figure III-35.

The operating and maintenance costs are estimated to be 0.15 cents/KWH of electric output. This estimate includes the costs of parts replacement. Table III-15 lists the average life expectancy of principal engine components when operating on boiler grade fuels.

The basic period maintenance work on the engine is as follows:

Approximately every 1500 hours

- Checking and spray testing of the fuel valves.

Every 6000 to 8000 hours

- Withdrawal of pistons.

- Measuring cylinder liner wear.
- Checking and, if necessary, replacing piston rings.
- Cleaning of turbochargers.
- Checking of fuel pumps and resetting if necessary.

In addition to the above principal checks, the filters for fuel and luboil should be periodically cleaned.

The engine will have to be shut down for 300-450 hours per annum to carry out maintenance. A yearly plant utilization factor of about 95% can thus be achieved. Unscheduled outages could be 1.5 percent resulting in an overall plant utilization factor of about 93 percent.

Emissions

Exhaust emissions for low speed diesel engines have been estimated based upon data provided by Sulzer Brothers. The predicted emissions for current technology with petroleum boiler fuel are presented in Figure III-36. The estimated emissions for the advanced technologies using coal-derived boiler fuel and powdered coal are presented in Figure III-37. In preparing the powered coal a significant portion of the sulfur compounds would be removed in a flotation process. The assumption was made that such a process could reduce the sulfur content of the coal to one percent. The nitrogen oxide levels from low speed diesel engines are higher than the emission guidelines defined in Volume I.

Physical Characteristics

The low speed diesel-generator systems, exclusive of balance of plant, typically weigh about 100 pounds per kilowatt and occupy about 0.4 square feet per kilowatt. No significant variation is anticipated between current practice and advanced technologies in commercial service in 1985-2000.

Cogeneration Applicability

The low speed diesel-generator is considered a promising candidate for cogeneration. It offers operational flexibility and is able to operate very efficiently over a wide range of output levels and to respond rapidly to change in demand while consuming boiler grade oil or even powdered coal. The diesel-generator can be grid connected or operated independently. Since diesel engines are modular, additions to meet increasing plant requirements are generally simple. The principal physical drawback to low speed diesel cogeneration applications is the relatively high level of exhaust emissions which can be of significant deterent in some areas.

Future Developments

Low speed diesel engines have demonstrated the efficiency, fuel flexibility, and durability suitable for cogeneration applications. However, to achieve extensive cogeneration application in the 1985-2000 time period, the following technical developments would be beneficial:

1. Development of high temperature water cooling systems.
2. Development of combustion systems which produce low levels of exhaust pollutants.
3. Development of engines able to operate with coal derived fuels with low levels of pollutants.
4. Development of durable powdered coal diesel engine systems.
5. Development of heat recovery equipment and cogeneration controls.

HIGH SPEED DIESEL GENERATORS

Conversion System Description

The advanced high speed diesel engine data used in this study were provided by the Cummins Engine Company, Incorporated of Columbus, Indiana. The current

technology data were provided by their subsidiary, the Cummins Cogeneration Company of New York.

The high speed diesel prime mover is a turbocharged, multi-cylinder, four stroke engine typically operating at 1800 rpm. The current technology engine is cooled by water circulated through a water jacket. Heat is recovered from the cooling water and also from the hot engine exhaust gases. This arrangement is shown schematically in Figure III-38 along with representative temperatures. Two current technology design options were included: energy conversion system number 3 design option number 1 is representative of power plants in the 400 to 1000 kilowatt size range. Design option number 2 is representative of engines in the 1 to 1/2 megawatt size range. Table III-16 presents the major design parameters for each of these design options.

A significant research effort for high speed diesel engines is aimed towards "adiabatic" power plants which do not have cooling water systems. The use of gas bearings and the application of ceramic parts in high temperature areas are the basis of such designs. Figure III-39 illustrates the application of ceramic parts on the surfaces of the pistons, cylinder walls, and valves which are exposed to high temperature products of combustion. The adiabatic configuration has reached the single cylinder test stage and complete engines based on this technology are expected to be commercially available in the 1985-2000 period. A simplified schematic of a high speed diesel engine cogeneration configuration and representative temperatures are included in Figure III-40.

Current technology high speed diesel engines use petroleum distillate fuel. Advanced technology power plants are expected to be able to operate on coal-derived distillate fuels corresponding to number 2 oil.

Performance Characteristics

The design point energy balance for the current technology high speed diesel-generators is presented in Figure III-41. Energy conversion system number 3, design option number 1 applies to a 400 kilowatt electrical output and design option number 2 is for an engine with 1000 kilowatt output.

The advanced design performance is also shown in Figure III-41. In this case a single design point was selected. The performance variation with size is expected to be small over the range from 0.4 to 1.5 megawatts, since parasitic losses associated with the oil lubrication and water cooling systems have been eliminated. The advanced technology is projected to be very efficient with estimated specific fuel consumption of 0.28 pounds per horsepower hour. Limiting the number of cylinders in the engine to 18 limits the maximum output for advanced high speed diesel engines to about 1 1/2 megawatts.

High speed diesel-generators operate at high efficiency from 25 percent to rated electrical output. The off-design performance of current technology high speed diesel-generators is included in Figure III-42 and the corresponding projected data for advanced technology is presented in Figure III-43.

Estimated Costs

The estimated costs of the high speed diesel engines were provided by the Cummins Engine Company and the Cummins Cogeneration Company. The cost estimates are summarized in Table III-17. The advanced technology system eliminates water cooling and oil lubrication components but these cost savings were judged to be comparable to the cost increases associated with the ceramic parts and the gas bearings. The installation costs were also provided by Cummins Cogeneration Company.

Typical maintenance and overhaul schedules for high speed diesel engines are presented in Table III-18. The estimated operating and maintenance costs were from 0.7 to 1.5 cents kWh. The value 0.7 cents/kWh was used in the study. The overall availability of the high speed diesel engines is estimated to be 96 percent which is due to 2 percent scheduled and 2 percent unscheduled outage.

Emissions

Estimated exhaust emissions for high speed diesel engines are presented in Figure III-45. As with the low speed diesel engines, the nitrogen oxide levels are higher than the guidelines defined in Volume I.

Physical Characteristics

The high speed diesel-generator, exclusive of balance of plant equipment, typically occupies about 1.5 square feet per kilowatt and weighs about 20 pounds per kilowatt.

Cogeneration Applicability

High speed diesel-generators have been the principal prime movers in "total energy" systems which are cogeneration systems in commercial and residential buildings. They could also be used in industrial cogeneration applications except for their size limitations. With individual units limited to about 1 1/2 megawatts, high speed diesel installations are limited to about 10 or 15 megawatts. The advanced high speed diesel engines operate at very high efficiency over a wide range of output levels and respond rapidly to changes in demand. They can be grid connected or operate independently. The engines are modular and the number of units can be increased to accommodate increased industrial process needs.

High speed diesel-generators can operate on diesel fuel (equivalent to Number 2 oil) or natural gas. Future engines will be able to use coal-derived distillate fuels. The high speed diesel engines emit relatively high levels of pollutants, particularly nitrogen oxides which can be a significant deterent in some areas.

Future Developments

High speed diesel engines have demonstrated the efficiency and operational flexibility for many cogeneration applications. To achieve the widest acceptance practical in the 1985-2000 period, the following technical developments would be desirable:

1. Development of the "adiabatic" engine to achieve required levels of cost and durability to provide commercially competitive equipment. This type of engine includes ceramic high temperature components and gas bearings.

2. Development of combustion systems which produce low levels of exhaust pollutants while maintaining or improving efficiency.

3. Development of engines able to operate with coal derived distillate fuels with low levels of exhaust pollutants.

4. Development of heat recovery equipment and controls.

TABLE III-12

LOW SPEED DIESEL UNIT CHARACTERISTICS
CURRENT/ADVANCED TECHNOLOGY

Type	2 Cycle Turbocharged
Size	8.1 to 28.6 MW
Bore	26.8 to 35.4 in.
Stroke	49.2 to 61.0 in.
Speed	150 to 120 rpm
Piston Speed	20 to 21 ft/sec
Firing Pressure	1310 psi
BMEP	179.5 psi
Mechanical Output	1900 to 3320 HP/Cylinder
Electrical Output	1350 to 2380 kW/Cylinder
Water Jacket Temp	158°F/266°F
Turbine Exhaust Temp	540°F
Fuel Type	Petroleum Residual/ Coal Derived Boiler Fuel/Coal
Sulfur Content	<1%
Electrical Generator Efficiency	96%

TABLE III-13

LOW SPEED DIESEL ENGINE COOLING REQUIREMENTS

Cylinder	860 Btu/BHP - Hr
Piston	240
Lubricating Oil	50
Charge Air	600
Total	1,750 Btu/BHP - Hr.

TABLE III-14
LOW SPEED DIESEL ENGINE SCHEMATIC DIAGRAM

NOMENCLATURE

Description of fluid streams

Primary streams

A = Air
E = Exhaust gas
C = Cylinder cooling water
P = Piston cooling water
T = Turbine cooling water
O = Oil

Secondary streams

H = High pressure steam or condensate
L = Low pressure steam or condensate
W = Hot water
R = Raw water

Description of equipment

Engine and ancillary equipment

a = Engine
b = Turbocharger
d = Compressor
e = Exhaust gas turbine
f = Piston cooling deairation and settling tank
g = High-pressure steam separation drum

Primary heat exchangers

h = Exhaust boiler
i = Air cooler
k = Low-pressure steam generator
m = Low-pressure steam condensate cooler
n = Hot water heater
p = Oil cooler

Secondary heat exchangers

q = High-pressure steam heat exchanger
r = Low-pressure steam heat exchanger
t = Hot water heat exchanger

Definition of flow direction:

1 = Flow entrance
2 = Flow exit

SYMBOLS

	=	AIR FILTER
	=	COOLING SYSTEM
	=	AIR - WATER HEAT EXCHANGER
	=	WATER - WATER HEAT EXCHANGER
	=	HEAT UTILIZATION
	=	STEAM - WATER SEPARATOR
	=	AIR - WATER SEPARATOR
	=	PUMP
	=	VALVE OR ADJUSTABLE THROTTLE

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TABLE III-15
LOW SPEED DIESEL - GENERATOR
PRINCIPAL COMPONENT REPLACEMENT SCHEDULE

Component	Replacement (Hours)
Piston Cooling Tube Seal	6,000
Fuel Injection Valve	10,000
Piston Ring	12,000
Fuel Injection Pump Valve	20,000
Cross Head Bearing	30,000
Cylinder Liner	45,000
Piston Skirt	50,000
Main Bearing	50,000
Cam Shaft Bearing	70,000
Cranking Gear	90,000

TABLE III-16
HIGH SPEED DIESEL ENGINE - GENERATOR CHARACTERISTICS
PRESENT TECHNOLOGY

	Design No. 1	Design No. 2
Plant Size Range	0.4 to 1 MW	1.0 to 1.5 MW
Unit Characteristics		
Type	4 Cycle Turbocharged Aftercooled	4 Cycle Turbocharged Aftercooled
Bore	5.50 inch	6.25 inch
Stroke	5.50 inch	6.25 inch
Speed	1800 rpm	1800 rpm
Piston Speed	27.5 feet/second	31.25 feet/second
Firing Pressure	2200 psi	2200 psi
Compression Ratio	14.5 to 1	14.5 to 1
Compression Pressure	1600 psi	1600 psi
BMEP	196 psi	196 psi
Mechanical Output	70 HP/cylinder	100 HP/cylinder
Electrical Output	49.1 kW/cylinder	70.1 kw/cylinder
Water Jacket Temp	200°F	200°F
Turbine Exhaust Temp	1000°F	900°F
Fuel Type	No. 2 Oil	No. 2 Oil
Sulfur Content	1% Max	1% Max
Fuel Consumption	10,200 BTU/kWh	9800 Btu/kWh
Electrical Generator Efficiency	94%	94%

TABLE III-17

HIGH SPEED DIESEL ENGINE-GENERATOR COST ESTIMATES

Dollars per Kilowatt

Component	Current Technology	Advanced Technology
Engine	121	129
Generator	24	24
Heat Recovery	18	22
Installation	93	85

TABLE III-18

HIGH SPEED DIESEL ENGINE

GENERATOR MAINTENANCE SCHEDULE

Scheduled Maintenance	Occurrence (Hours)	Duration (Hours)
Oil Change	600	2
Checkout	1500	5
Checkout	4500	8
OVERHAULS		
Minor	15,000	12*
Major	30,000	12*

* Retrofit Unit Assumed

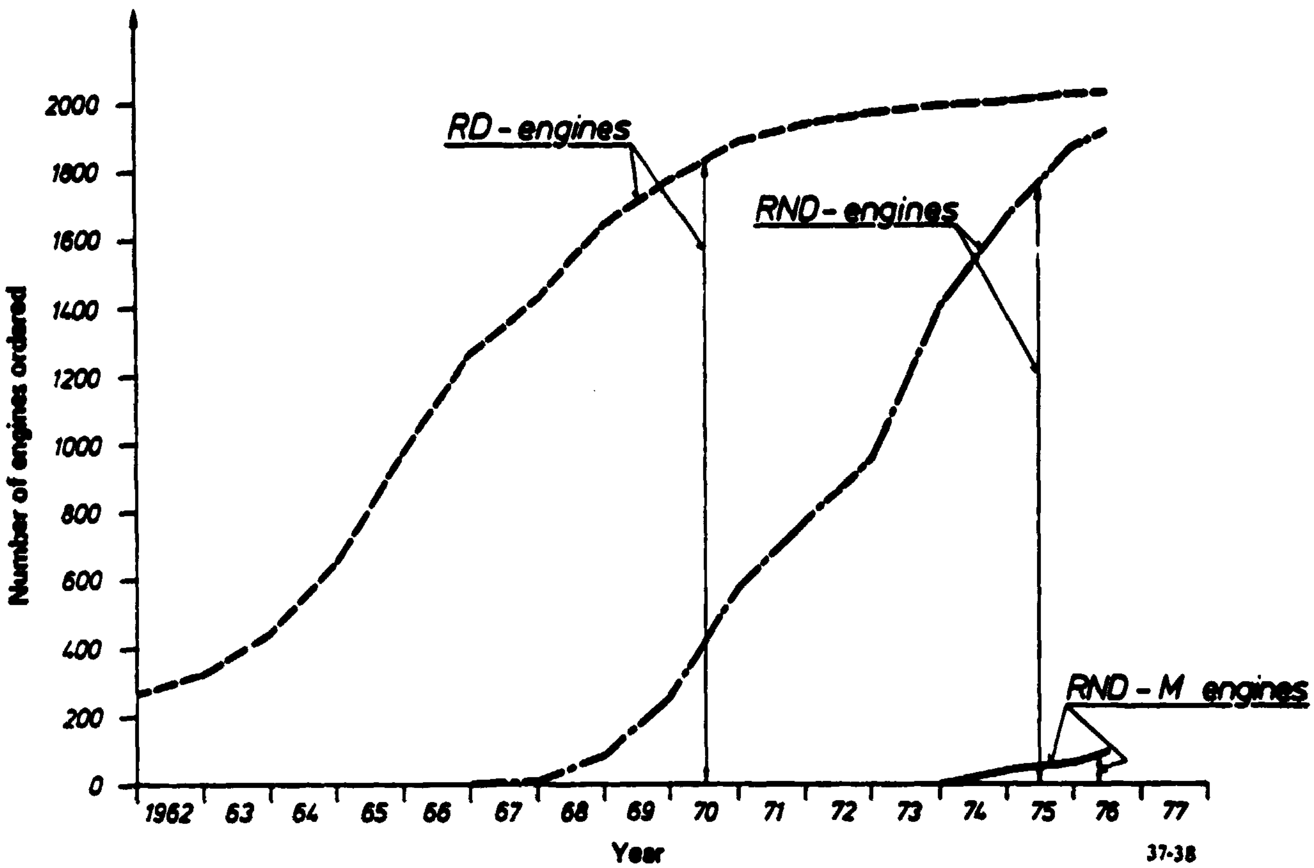


Figure III-20. Sulzer Low-Speed Diesel Engines Ordered

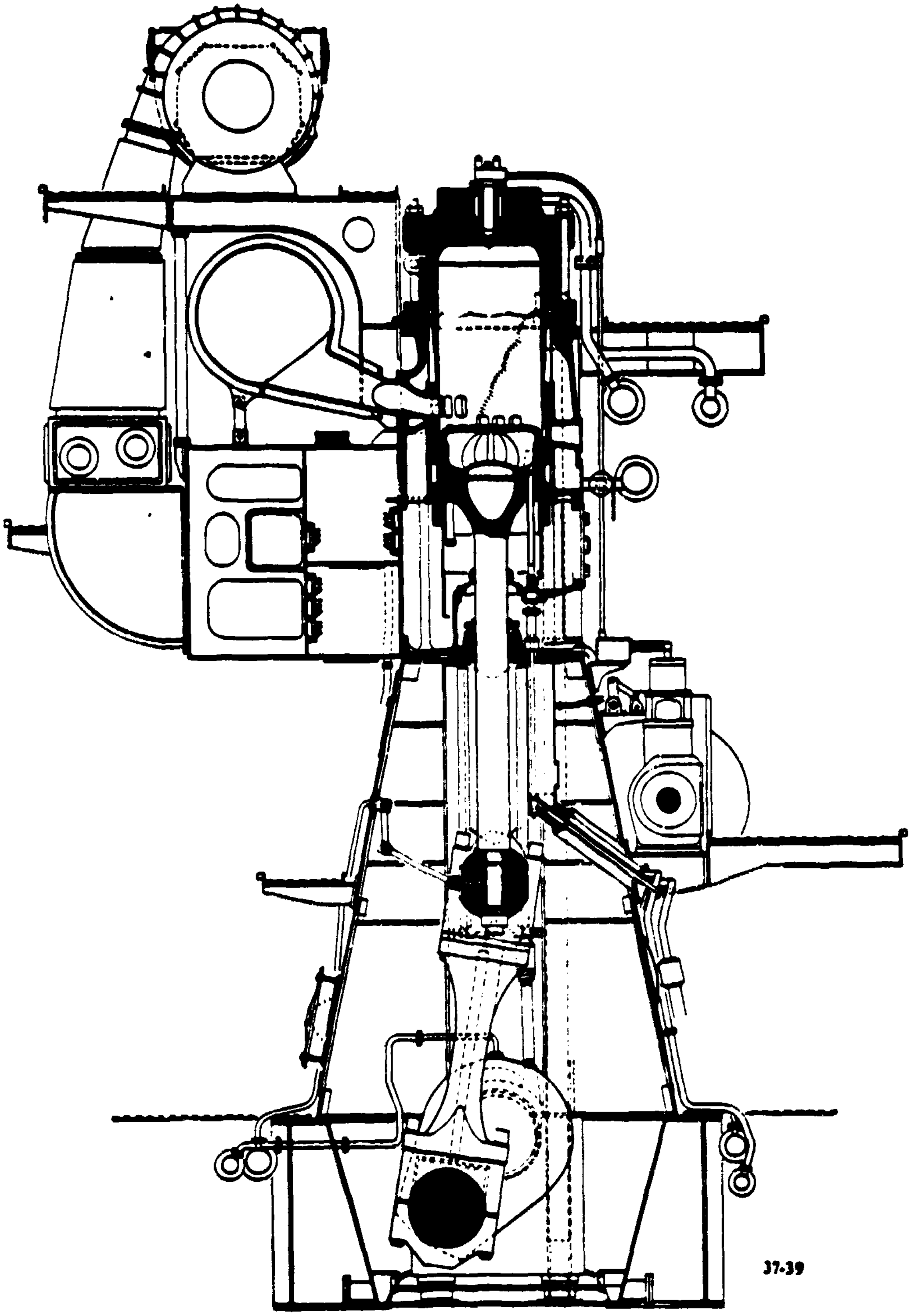
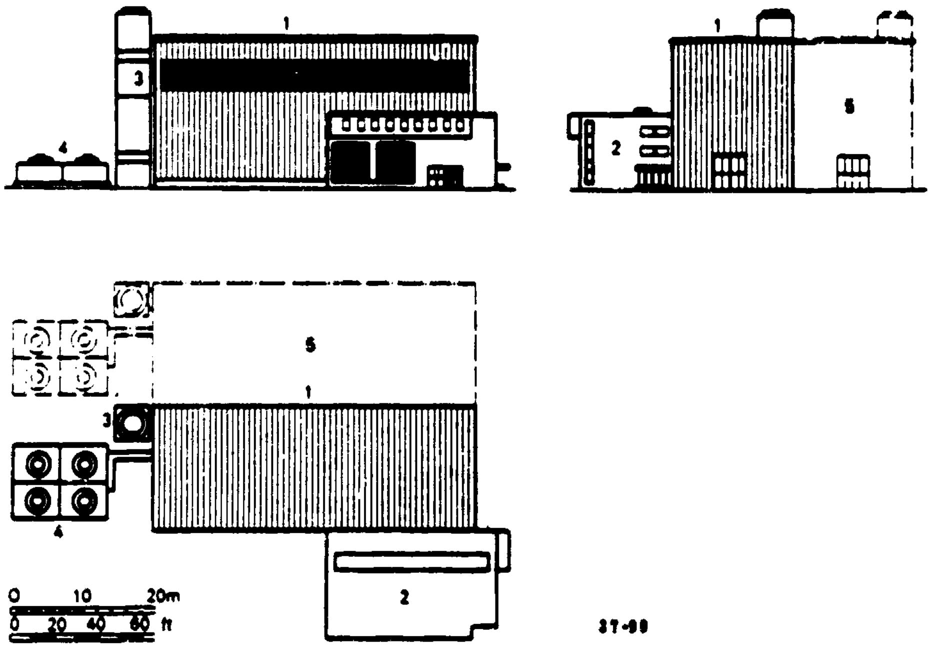


Figure III-21. Low Speed Diesel Engine Cross-Section



1 Machinery hall
2 Auxiliary building
3 Exhaust chimney with silencers
4 Cooling towers
5 Possible extension of machinery hall

Figure III-22. Low Speed Diesel Electric Generating Station - External View

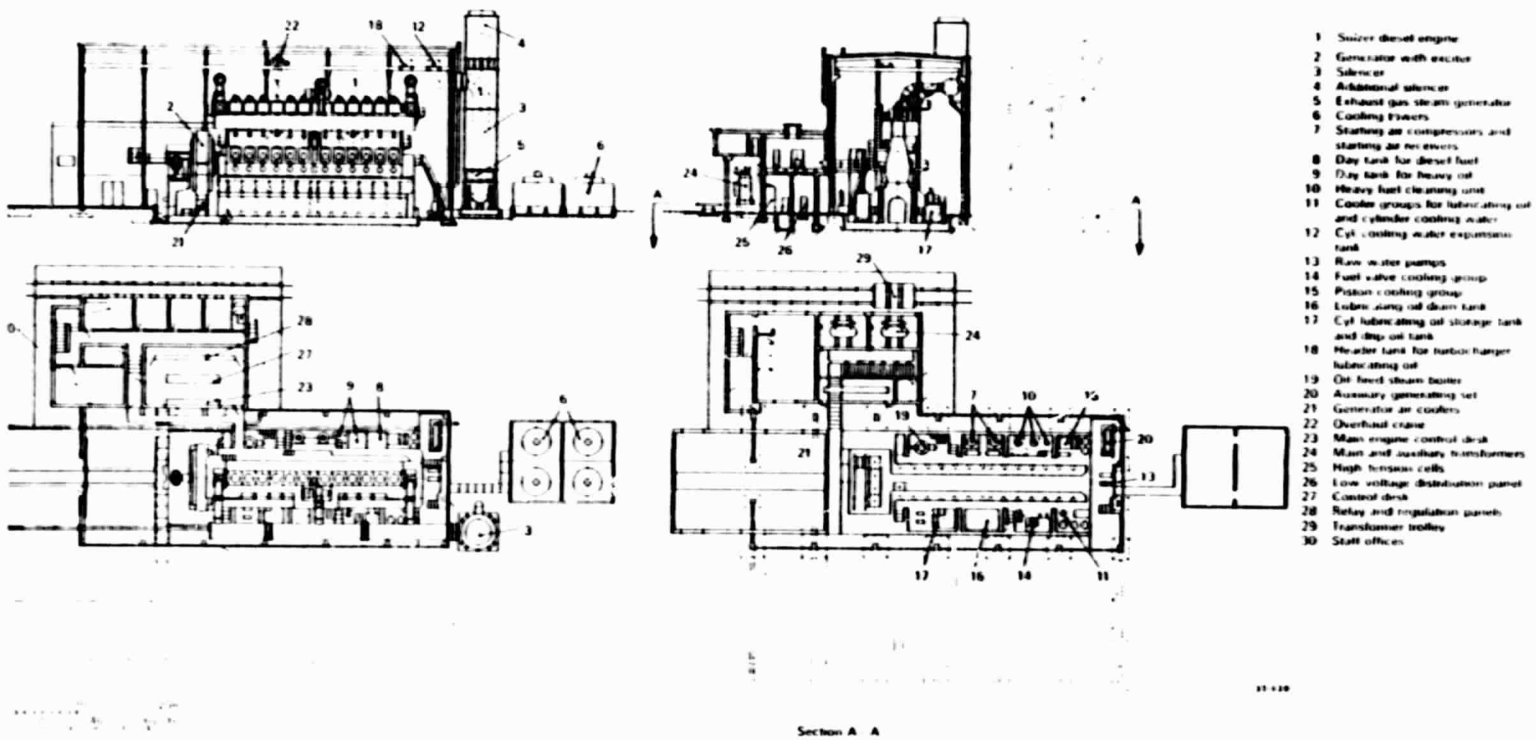


Figure III-23. Low Speed Diesel Generator - General Plant Arrangement

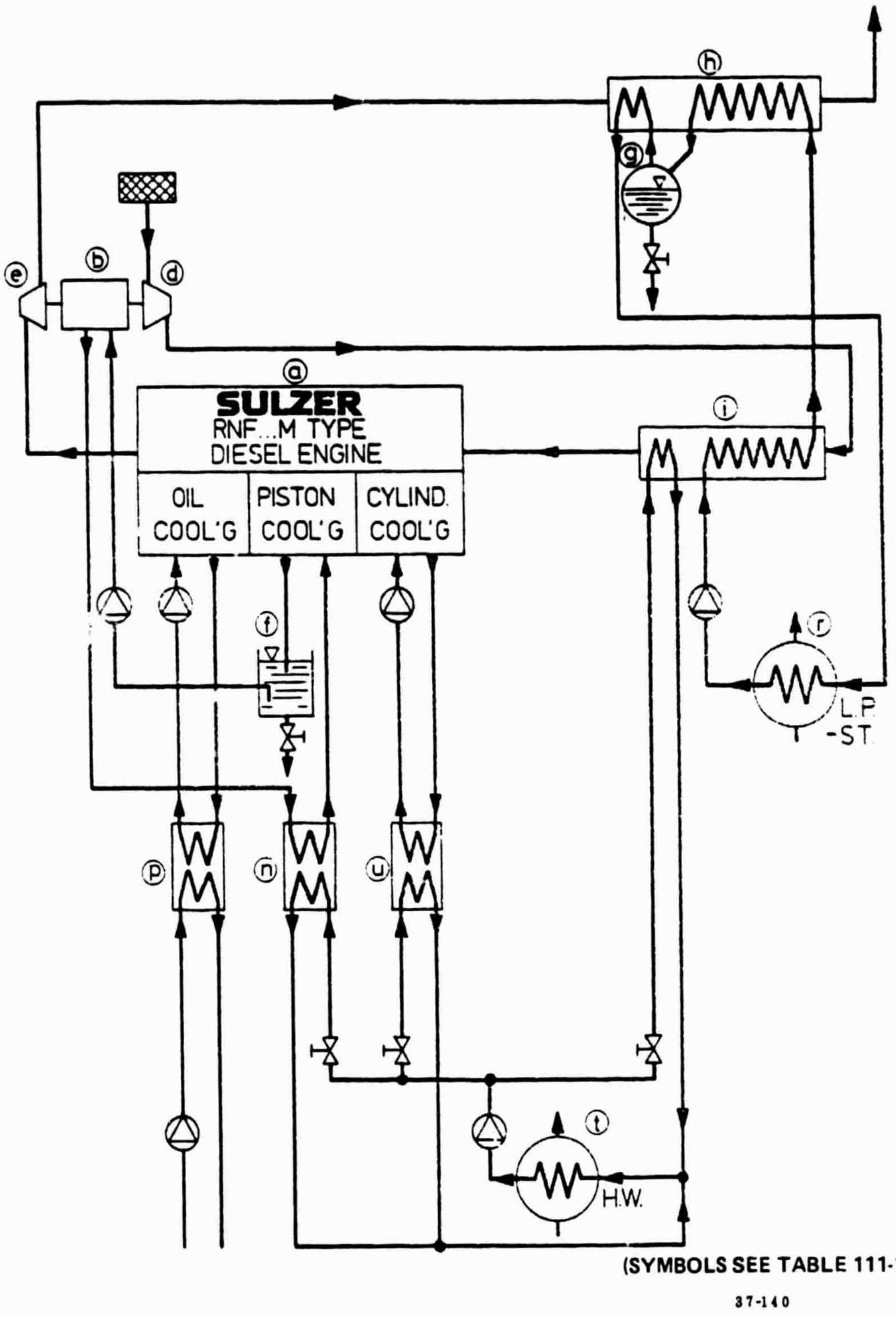


Figure III-24. Schematic Diagram of Heat Recovery Arrangement, Current Technology Low Speed Diesel-Generator

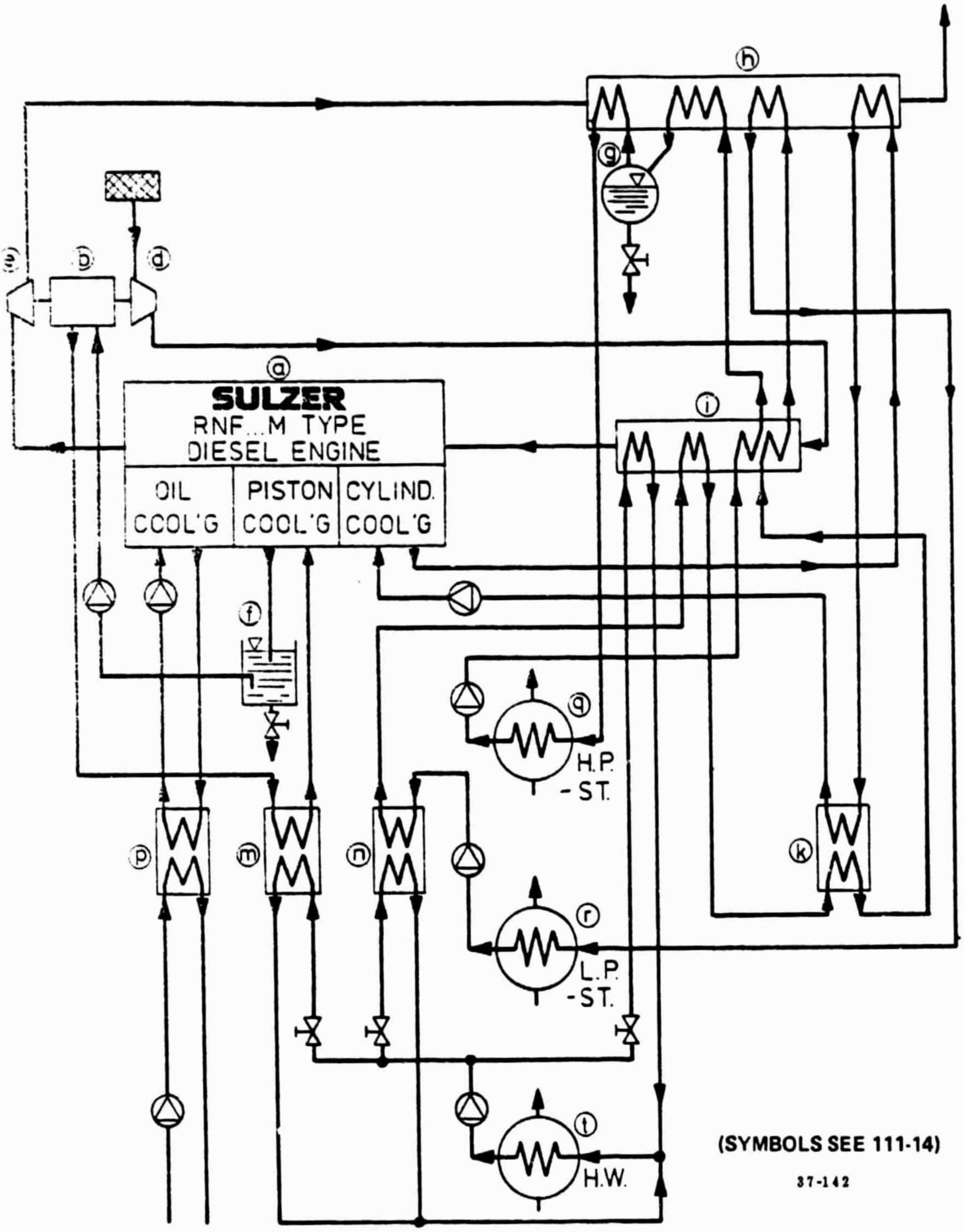


Figure III-25. Schematic Diagram of Heat Recovery Arrangement, Advanced Technology Low Speed Diesel-Generator

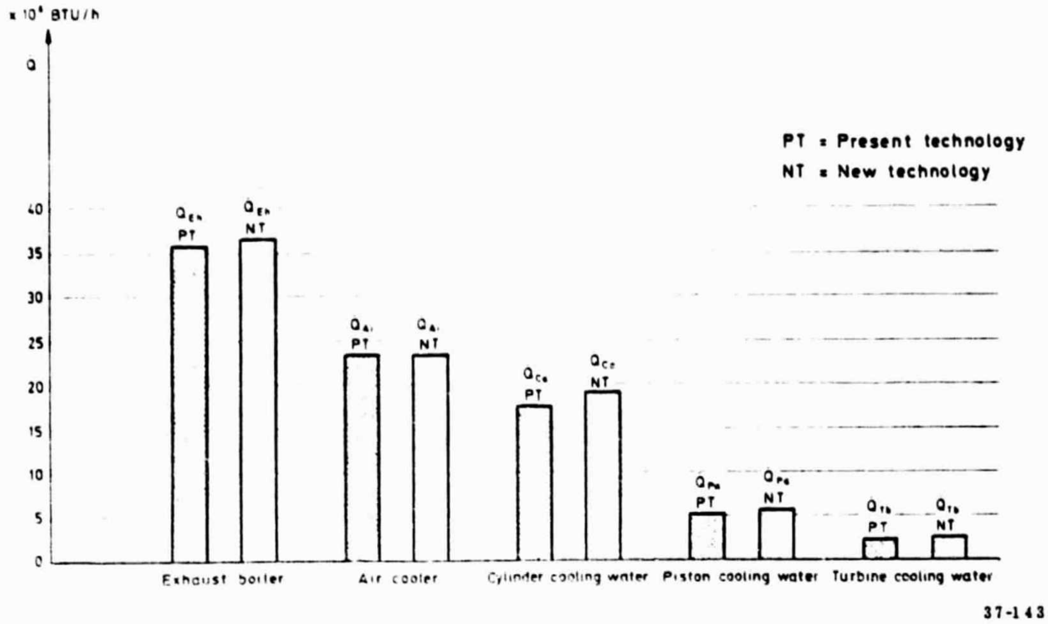


Figure III-26. Component Heat Recovery from 28.5 MW Low Speed Diesel Engine - Generator, 90% Output

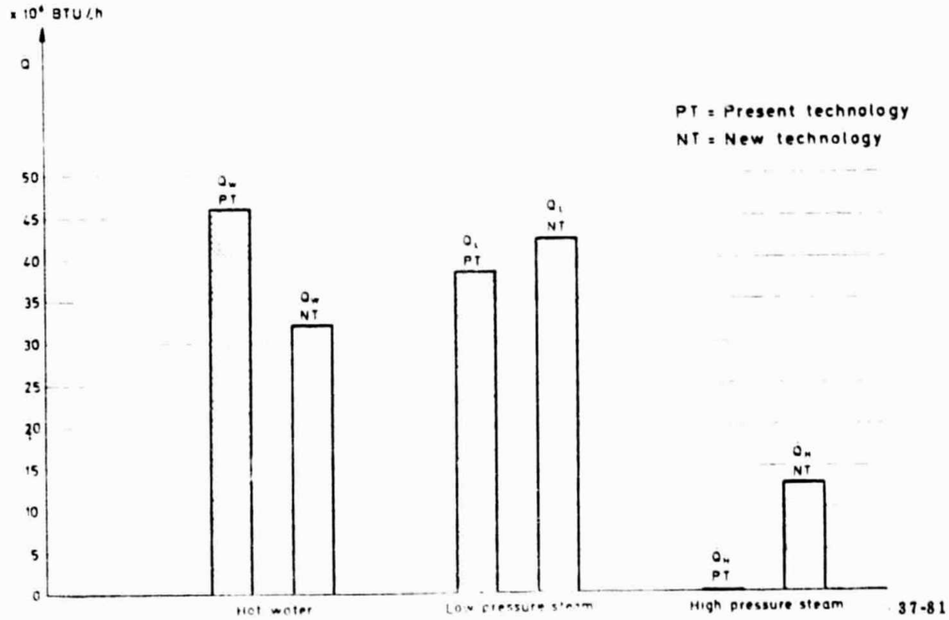


Figure III-27. Heat Recovery from 28.5 MW Low Speed Diesel Engine - Generator, 90% Output

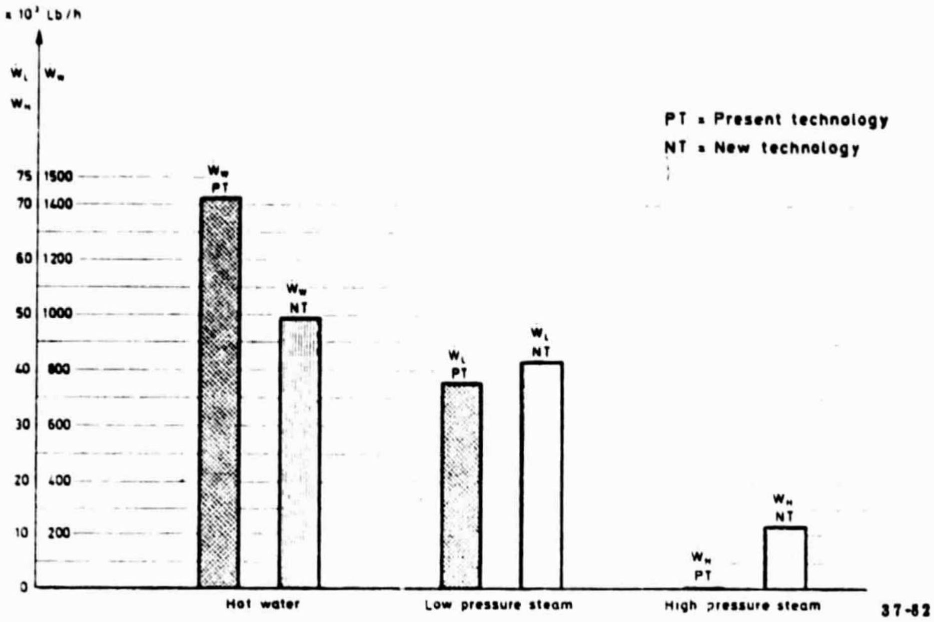


Figure III-28. 28.5 MW Low Speed Diesel Engine, 90% Output

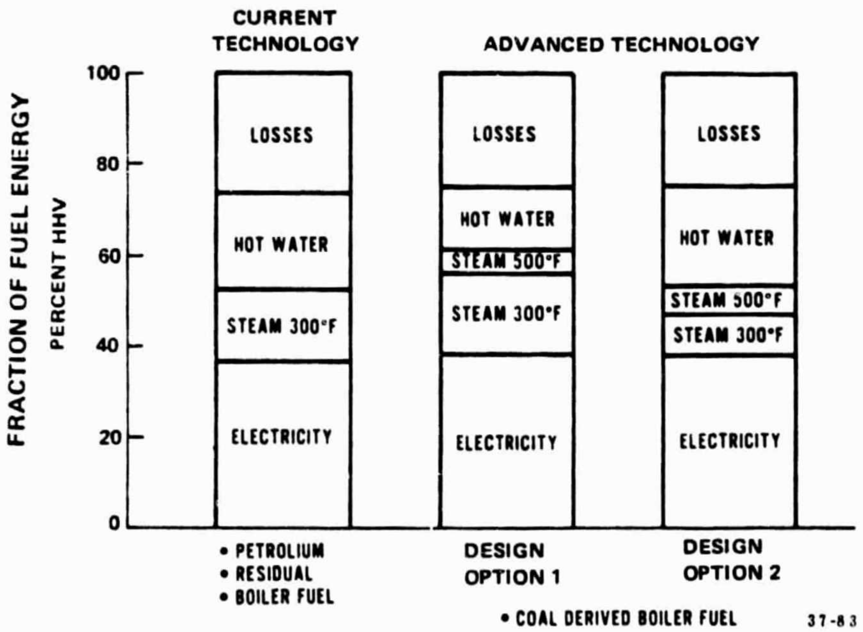


Figure III-29. Low Speed Diesel Energy Balance

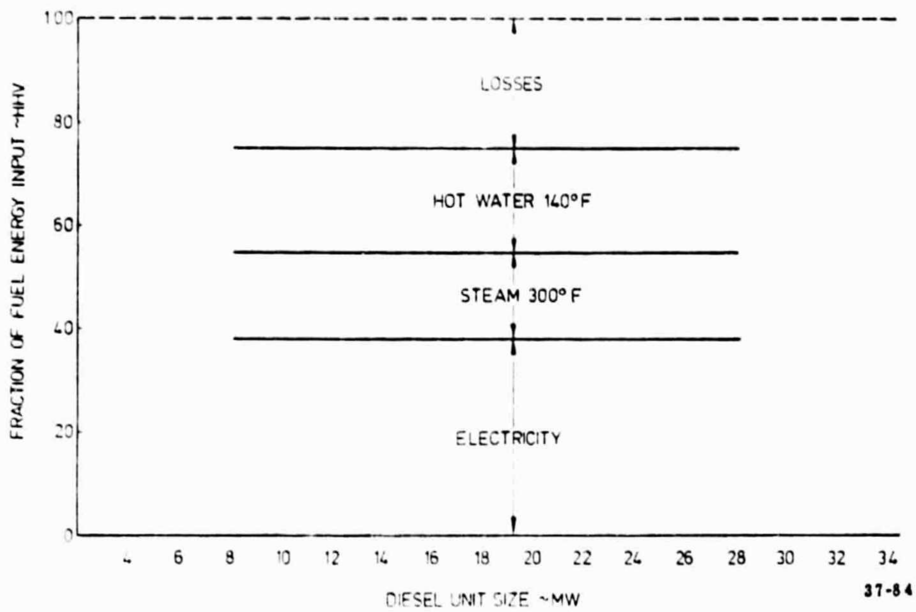


Figure III-30. Current Technology Low Speed Diesel Generator Unit - Energy Balance

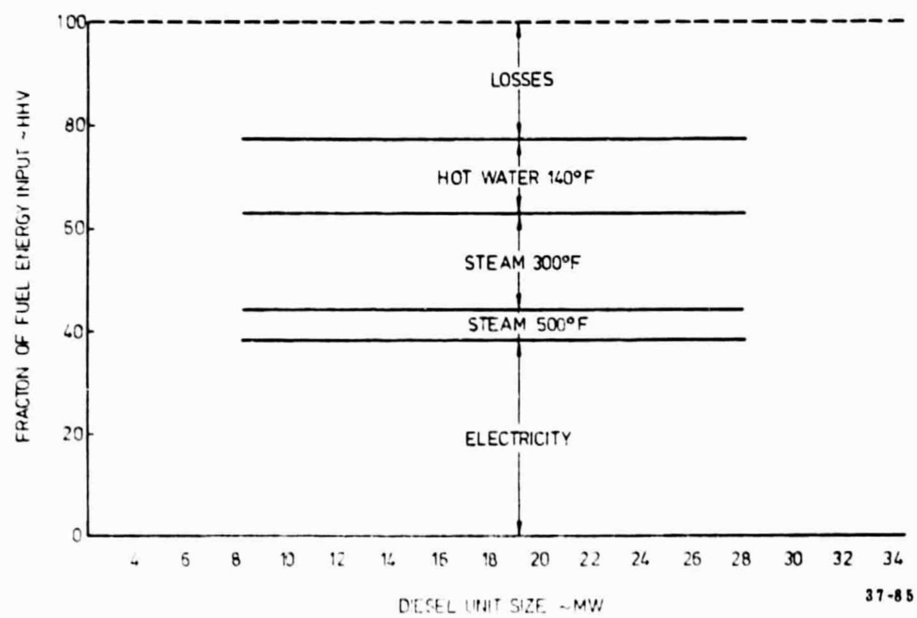


Figure III-31. Advanced Technology Low Speed Diesel Generator Plant - Energy Balance

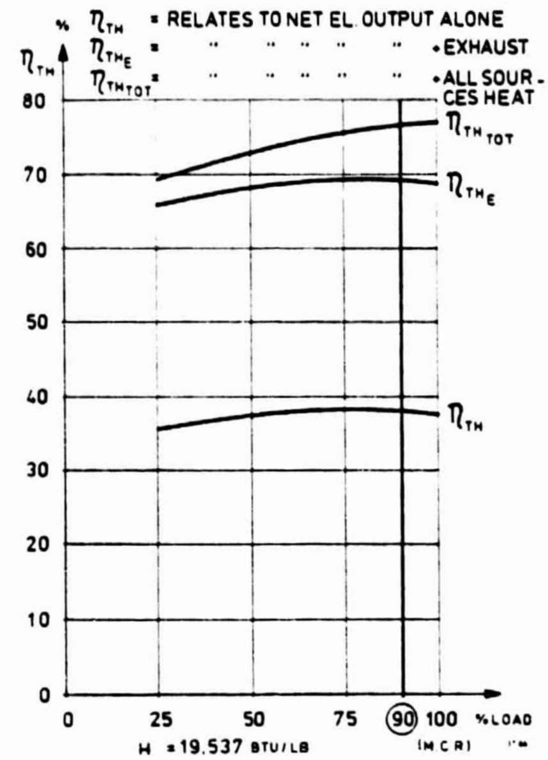


Figure III-32. Advanced Technology Low Speed Diesel-Generator Off-Design Performance

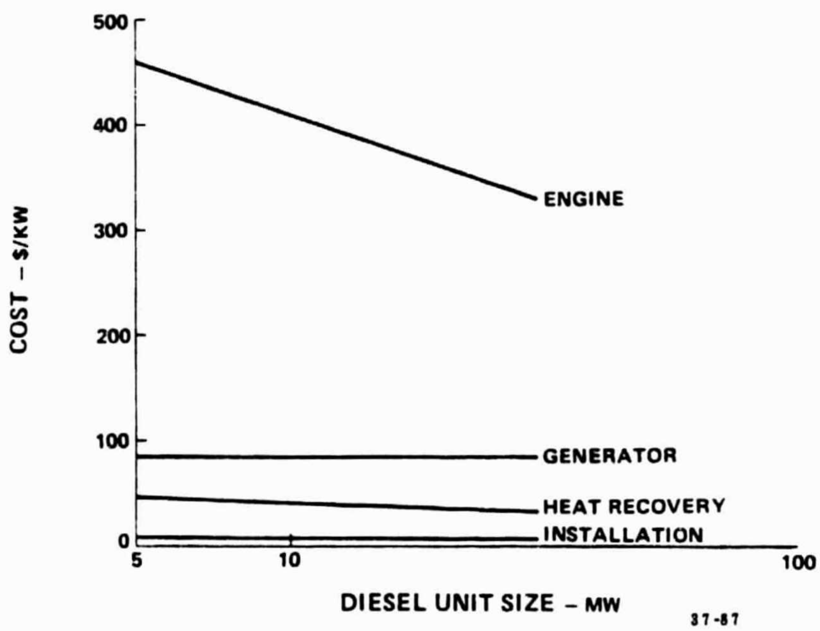


Figure III-33. Current Technology Low Speed Diesel Engine - Generator Estimated Cost

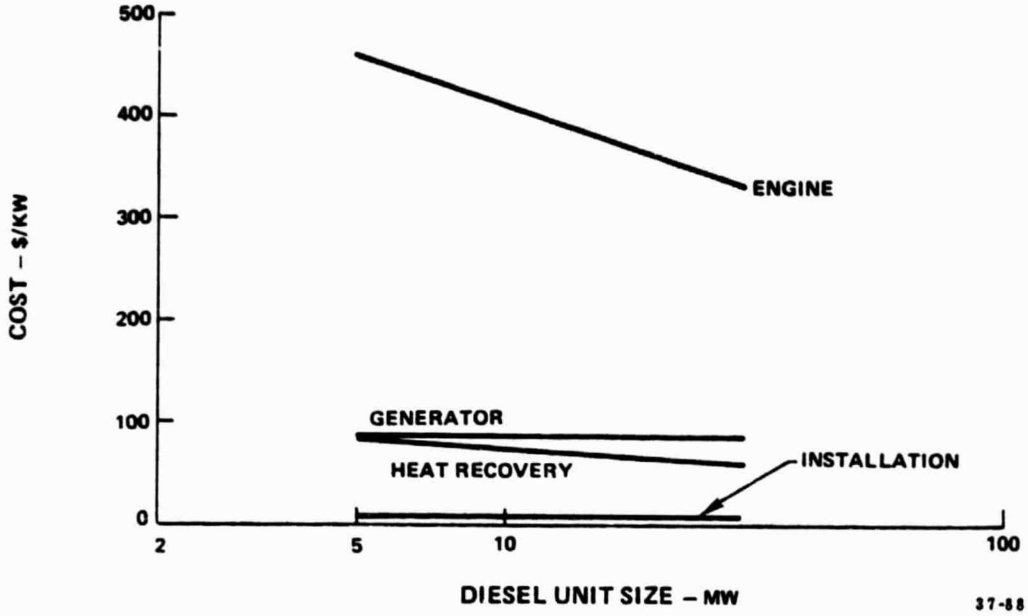


Figure III-34. Advanced Technology Low Speed Diesel Engine - Generator Estimated Cost

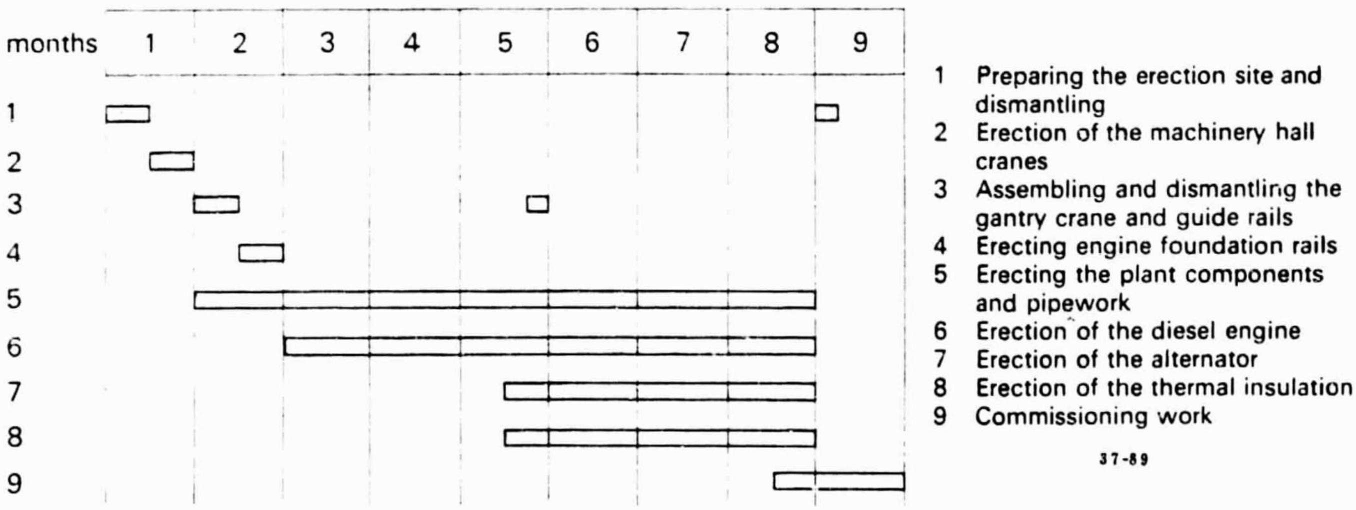


Figure III-35. Low Speed Diesel Engine - Generator - Installation Schedule

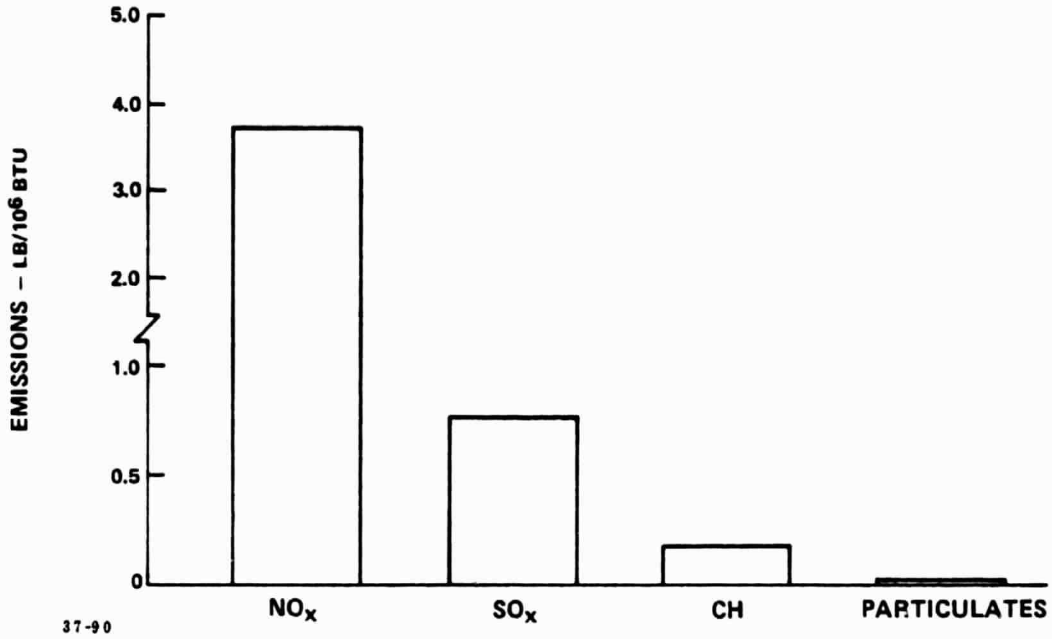


Figure III-36. Current Technology Low Speed Diesel Engine - Generator Emissions Petroleum Boiler Fuel

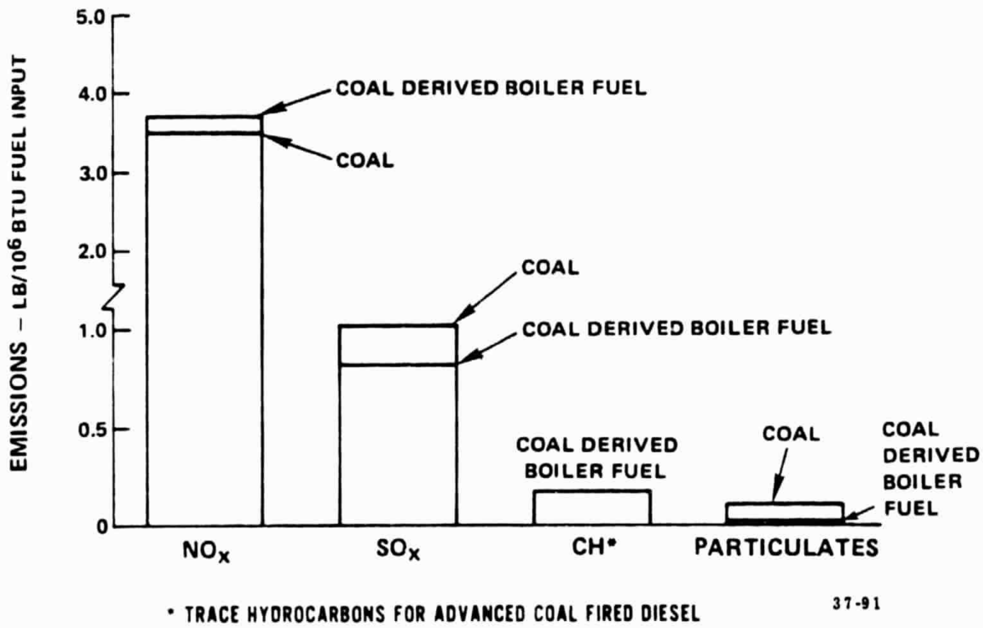


Figure III-37. Advanced Technology Low Speed Diesel Engine - Generator Emissions (Coal Flotation Sulfur Removal System Employed)

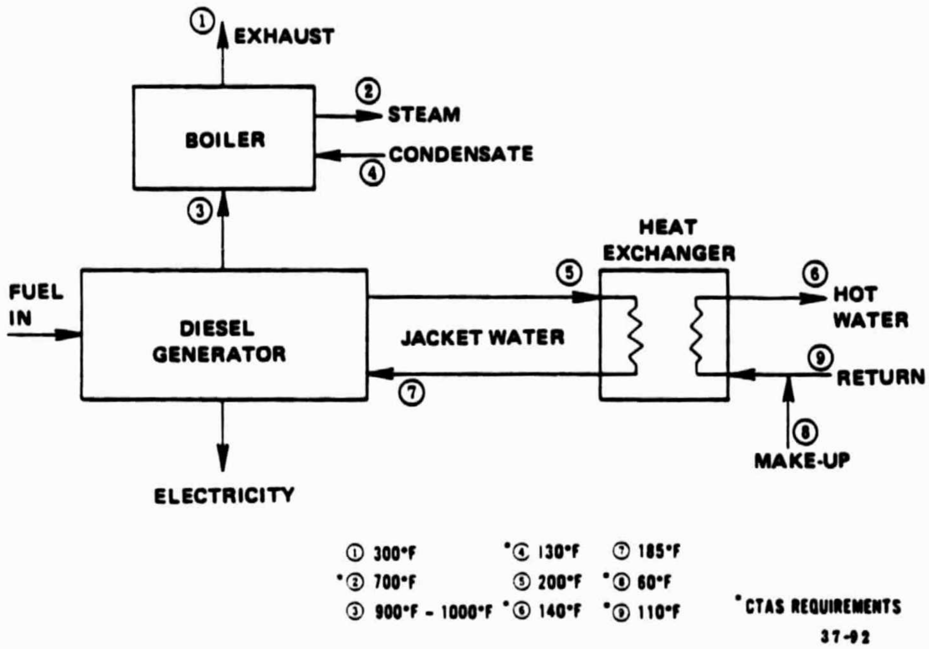


Figure III-38. High Speed Diesel Engine - Generator - Present Technology

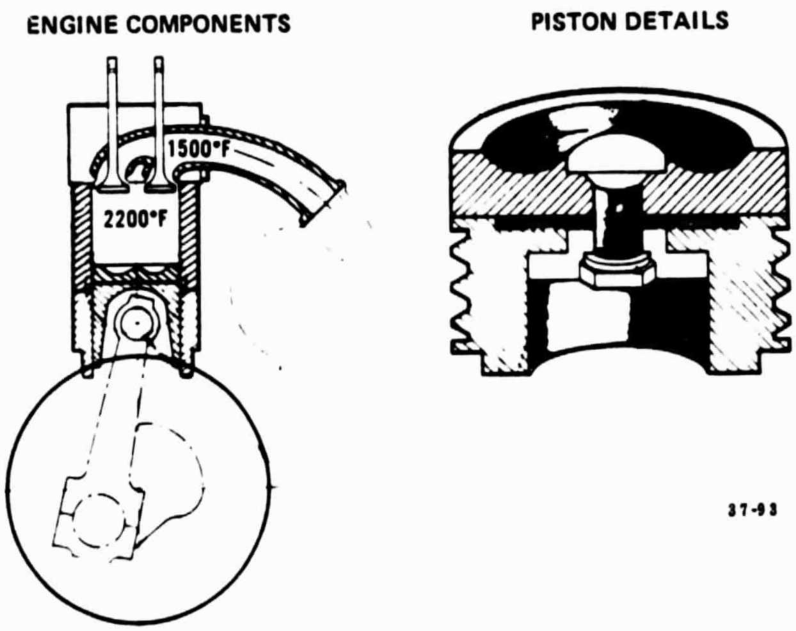


Figure III-39. Advanced Adiabatic Diesel - Ceramic Elements Cross-Hatched

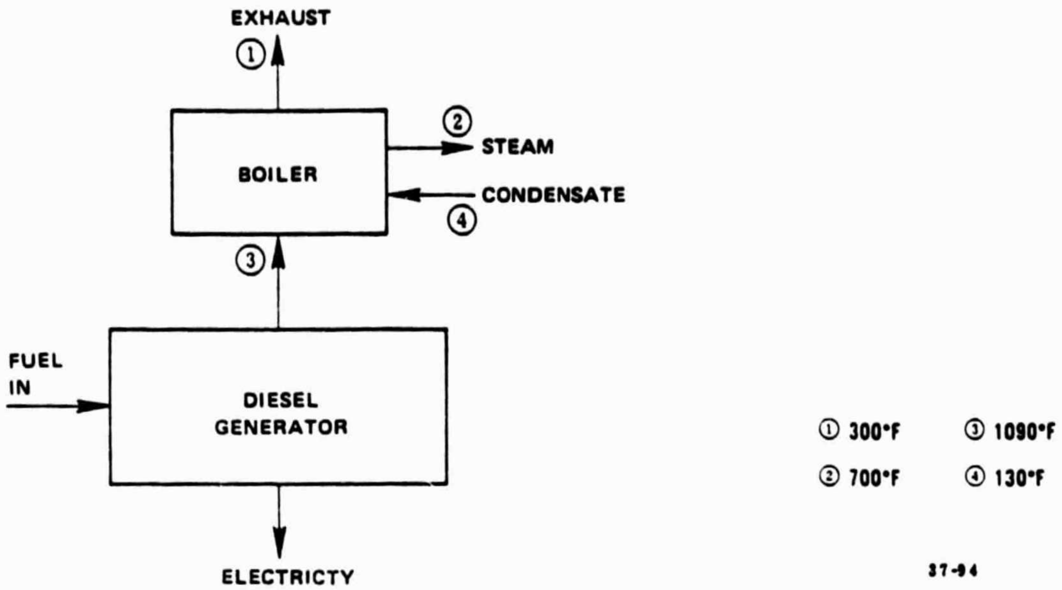


Figure III-40. High Speed Diesel Engine - Generator - Future Technology

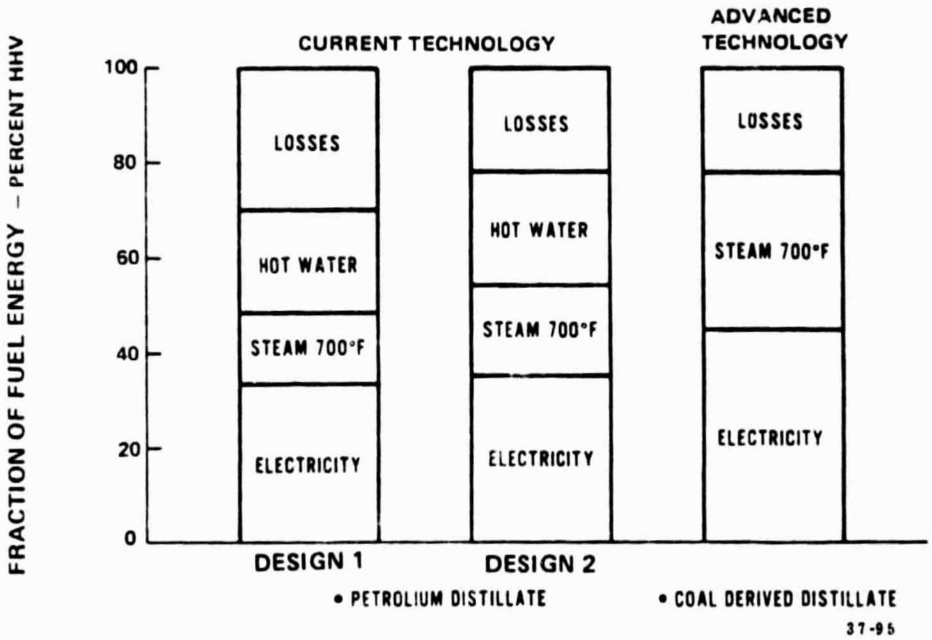


Figure III-41. High Speed Diesel Engine - Generator - Energy Balance

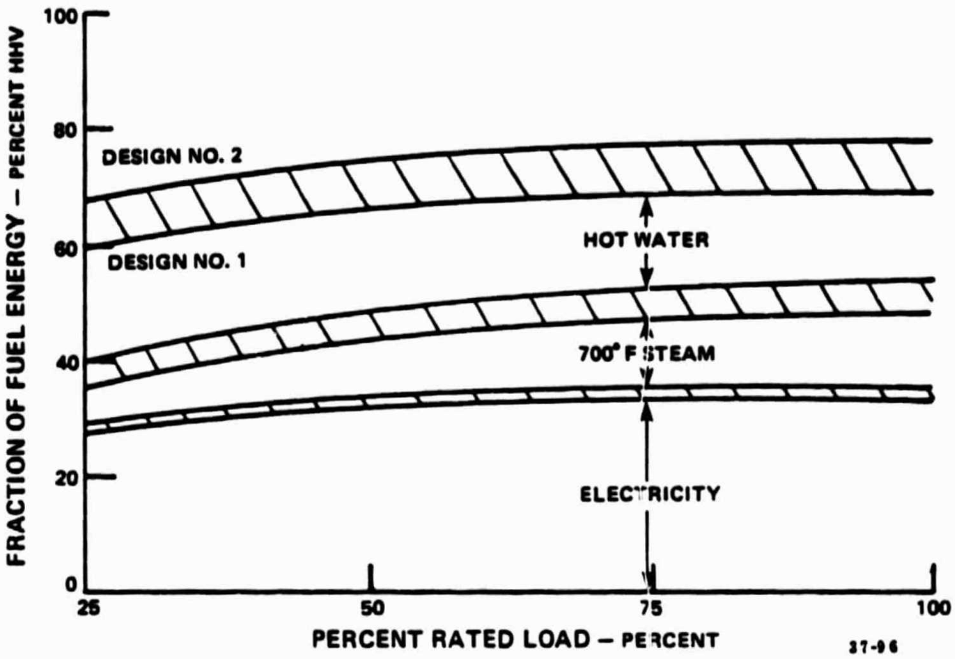


Figure III-42. High Speed Diesel Engine - Generator - Off Design Performance - Current Technology

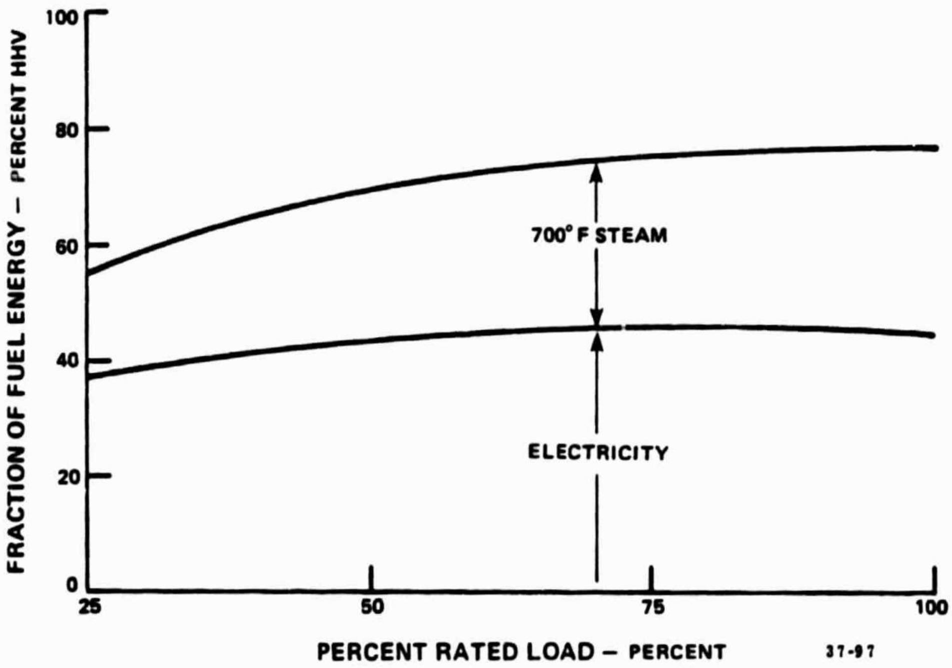


Figure III-43. High Speed Diesel Engine - Generator - Off - Design Performance - Advanced Technology

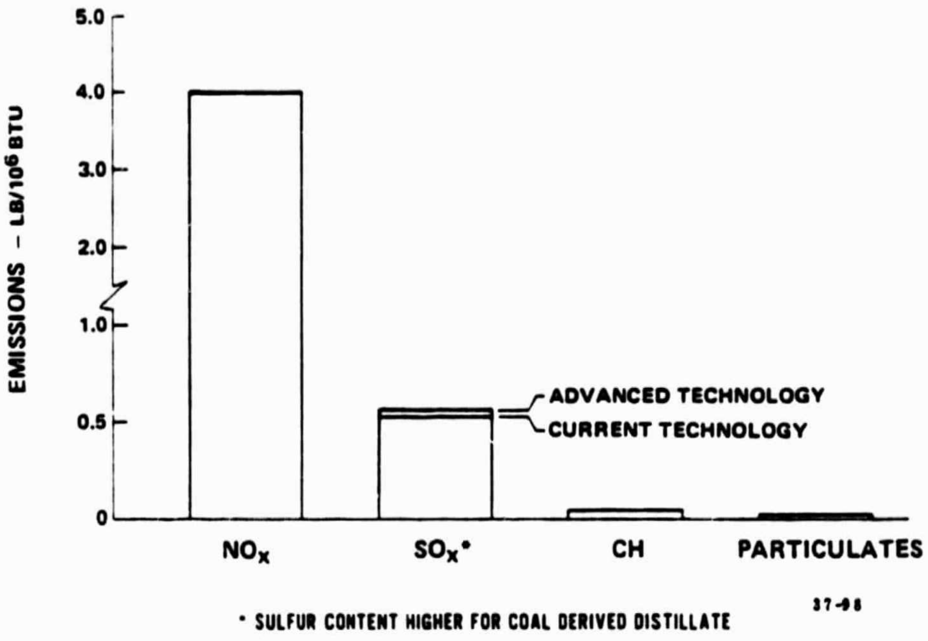


Figure III-44. High Speed Diesel Engine Estimated Emissions

GAS TURBINES

INTRODUCTION

Gas turbines, such as shown in Figure III-45, have been widely used in transportation, utility, and industrial applications. Many have operated in cogeneration applications such as the installation shown in Figure III-46. Other gas turbines have been coupled with heat recovery boilers and steam turbines and operated as combined cycle power plants. An installation of three gas turbines with heat recovery boilers is shown in Figure III-47. A single power plant is housed in the unit on the left and the turbine exhaust passes through the boiler in the center. In the housing on the right, two gas turbines drive a single generator in the center. The exhaust from each turbine passes through the bifurcated duct to the heat recovery boiler. Most gas turbines currently operate with natural gas or petroleum distillate fuels, but there are a number of installations using heavier industrial fuels.

An elementary schematic diagram of a gas turbine with heat recovery is included in Figure III-48. Air is drawn into the compressor, compressed, heated, and expanded through a turbine. The turbine drives the compressor and an electric generator. The turbine exhaust gas is typically about 1000°F and is the source of recovered heat for the industrial process.

There are a variety of gas turbines and a variety of ways they can be used in cogeneration applications. One variation is the method of heating the working fluid. With direct heating, fuel is injected into the compressor discharge air and burned. The hot air and products of combustion pass through the turbine. In most cases, about one quarter of the oxygen in the air is required for combustion. With indirect heating, fuel is burned with a separate air supply, and the resulting heat is transferred to the working fluid through a heat exchanger.

Figure III-49 presents simplified schematic diagrams of the principal means of gas turbine cogeneration. The first "simple" (A) system is described above. However,

there is a simple cogeneration configuration not shown whereby the turbine exhaust gases are used directly in the industrial process. This direct heat possibility may be limited by the industrial gas cleanliness requirements. An indirectly heated system has all of the oxygen in the air available for supplemental combustion, if required. The turbine exhaust, with indirect combustion, does not include products of combustion and can serve as clean hot gas for certain industrial processes.

If a large amount of heat is required by the industrial process, the turbine exhaust can be burned in a supplemental boiler, Figure III-49(B) since only one quarter of the oxygen in the working fluid is used within a power plant.

In the combined cycle, Figure III-49(C) the steam generated in heat recovery boiler is used to drive a steam turbine. Process steam can be extracted from the steam turbine or taken directly from the boiler, bypassing the turbine, as the needs of the industrial process dictate. A modification of the combined cycle is the steam injected gas turbine (D). In this case, some of the steam from the heat recovery boiler is injected upstream of the turbine. In effect, the gas turbine acts in the dual role of gas and steam turbine. Process steam is taken directly from the boiler. The closed cycle (E) is a logical system for cogeneration. Since the steam boiler removes most of the heat from the turbine exhaust, the amount of cooling required after the boiler for the compressor return is minimum.

Advanced coal fired gas turbine systems could provide four alternatives, Figure III-50. Atmospheric fluid bed coal combustion provides two possibilities. In the simplest case, (A) the air from the compressor discharge is piped through the combustion chamber where it is heated before entering the turbine. The fluid bed operates at 1500 to 1550°F which limits the turbine inlet temperature. The combustion air is provided by a separate blower which also provides the energy to maintain the coal and limestone fluidized condition.

Since the pressure of the gas turbine exhaust air is slightly above atmospheric, a portion of this stream can be used in the fluid bed combustion chamber, and the fluid bed blower can be eliminated, (B). Typically, a quarter of the turbine exhaust would be used in the combustion chamber and the rest would pass through a boiler to raise steam for the industrial process. If larger amounts of steam are needed by the industrial process, a larger amount of air can be bled to the combustion chamber with additional coal and limestone. A boiler would be added in the combustion chamber to accept the increased heat release.

The compressor discharge air can be used at pressure in a fluid bed combustion system, Figure III-50(C). Again, about a quarter of the compressor discharge air would be used in the pressurized fluid bed combustion, and the rest would be piped through the combustion chamber to receive heat. The products of combustion are cleaned and the solid particles removed and this gas, along with the heated air, feed the turbine and subsequent process heat recovery heat exchanger.

The last coal-fired system, (D) includes an on-site, entrained flow, air blown, coal gasification plant operating at 100 to 150 psi above compressor discharge pressure. The gasifier effluent is cleaned and introduced in the burner.

A number of gas turbine options were included in the study and these are summarized in Table III-19. The numbers in the table are the energy conversion system identification numbers used in the numerical analyses and with the data presented in Volume VI.

The following sections discuss each major gas turbine system including the system status, description, performance, capital costs, emissions, physical characteristics, operating and maintenance costs with references to other sections for comparisons.

First, simple cycles where the gas turbine exhaust is used for heat recovery are considered. These sections are then followed by injection of some of the steam raised by the exhaust heat into the combustor of the gas turbine to produce additional electricity. The last sections consider combined cycles where part of the recovered exhaust heat is used for additional electric generation.

Within the three groups of cycles (simple, combined and steam injection) the sections progress from direct combustion of clean oil tc direct combustion of heavy oils and coal-derived liquids. Direct processing of coal into a low BTU fuel gas is then discussed followed by direct combustion of coal in a pressurized fluidized bed. Then consideration of indirect combustion is discussed beginning with the open cycle followed by low temperature and high temperature closed cycles.

These sections on simple cycles completed, combined cycles are then discussed. Again, the pattern of fuel type is considered going from current clean oil, through coal derived liquids and ending with direct combustion of coal in a pressurized fluidized bed.

The options of steam-injected cycles included direct-fired heavy oil and direct and indirect combustion of coal.

TABLE III-19

GAS TURBINE ENERGY CONVERSION SYSTEM AND FUEL COMBINATIONS IDENTIFIED IN TABLE III-1

	PETROLEUM		FUEL COAL-DERIVED		COAL
CONVERSION SYSTEM	DISTILLATE	BOILER FUEL	DISTILLATE	BOILER FUEL	
Gas Turbine					
o Current Direct Fired	5				
o Direct Fired		12	13		14*, 15
o Indirect Fired					16
o Closed Cycle			17		18
Steam Injected Gas Turbine					
o Direct Fired		19	20		21
o Indirect Fired					22
Combined Cycle					
o Current Direct Fired	6				
o Direct Fired		23	24		25
o Indirect Fired					26

* Coal Gasification on Site

CURRENT GAS TURBINE DIRECT FIRED DISTILLATE FUEL

Current Status

A large number of gas turbine types and sizes are currently available. Some of these turbines have been available for many years, while others have been introduced very recently. A sampling of data representing the current state of the art for these current gas turbines is shown in Figures III-51, III-52, III-53 and III-54. These data are for peak power conditions. At base load conditions, the turbine inlet temperature would be about 100°F lower, and the other values would be correspondingly reduced.

It can be seen from Figure III-51, that some current gas turbines have peak turbine inlet temperatures of over 2100°F. Others, especially in smaller sizes, are about 300°F lower. The turbine exhaust temperature varies between 900°F and 1100°F regardless of size. Although the smaller engines operate at about 1800°F, they have lower overall pressure ratios and therefore have a lower temperature drop in the turbine.

Figure III-53 shows the overall pressure ratio as a function of size. Most of the engines below 10 MW have lower pressure ratios and higher heat rates than engines larger than 10 MW. This trend is due to the difficulty of making small parts with tight tolerances and leads to the relatively higher costs of complex engines in the smaller sizes.

Engines below 10 MW also have lower turbine inlet temperatures than the larger engines due to the difficulty of holding tolerances on the smaller parts which require complex cooling configurations to levels proportionate at larger parts.

Conversion System Description

Advanced gas turbines which could be available in the 1985-2000 period could burn heavy fuels such as residual petroleum oil, and heavy coal-derived liquids. These

power plants would be smaller for a given power and would have improved component performance.

From Figures III-51, III-52, III-53 and III-54 the base parameter levels were chosen. The data shows a range of power output between 600 kW and 100MW. 30 MW was chosen as the base size level and is representative of current commercial gas turbine offerings. Pressure ratios are seen to vary from 6 to 14:1 with a few special cases even higher. Base levels of 10:1, 12:1 and 14:1 were selected as being representative of current engines in the 30 MW range. The lower pressure ratios generally apply only to small engines. Turbine inlet temperatures for base load applications are relatively constant at 2000°F above 10 MW. Therefore, this value was assumed for the base parameter. The current technology gas turbine engines are also air-cooled. A current technology cycle is shown on Figure III-55 with typical parameter levels.

Performance Characteristics

The design point performance of system for the three base parameter pressure ratio levels is presented in Figure III-56 in terms of the fraction of fuel energy input based on the higher heating value of the fuel. Figure III-56 shows the electrical output, heat available in terms of steam and hot water and unrecoverable losses. The unrecoverable losses represent the combination of generator heat losses, heat of vaporization of the water in the exhaust system, heat losses due to radiation and convection, and residual heat exhausted to the atmosphere. The figure shows minor changes in overall energy utilization as a result of overall pressure ratio variation. The quantity and quality of the steam produced decline slightly with increasing pressure ratio. The decline is attributed to increased cycle efficiency which reduces the turbine exit temperature from 1020°F to 900°F as the pressure ratio increases from 10:1 to 14:1.

The off-design performance for current gas turbine engines, Figure III-57, indicates the typical increase in heat rate with reduced power output.

Estimated Costs

The cost estimate for the current gas turbine plus waste heat recovery units is shown in Figure III-58. These estimates presented for the three base parameter pressure ratios show only minor cost variations for the entire system. This is consistent with the performance results shown in Figure III-56 wherein the total system efficiency was seen to vary only a few percentage points. Figure III-58 was derived from the estimated cost of the gas turbine and generator Figure III-59, and the estimated heat recovery boiler costs, Figure III-60. Since the cost variation is so small for the various design options, a single system of costs was used in the computer analysis described in Volume V. The data used are summarized in Volume VI, Table VI-10.

The estimated operating and maintenance costs chargeable to the gas turbine are 2.5 mils/kWhr. This representative value was chosen as a result of examining the available literature in addition to manufacturers' data. Data from the literature, such as that reported to the Edision Electric Institute, is inconsistent but generally indicates a higher level of maintenance cost than manufacturers' data. The higher value was used for this study to account for less than optimum maintenance on the operator's part.

Emissions

The emission levels for current gas turbine technology are shown in Table III-20. The sulfur in the fuel is oxidized in the combustor and exhausted directly as sulfur dioxide. This is true of combustion systems where the sulfur and oxygen are present at high temperature and no sulfur absorbent or chemical treatment is present.

The current distillate fuel contains negligible amounts of nitrogen compounds. NOx is formed in the high temperature region of the gas turbine combustion system. This NOx, usually referred to as thermal NOx, is currently controlled to legal levels by water or steam injection to lower flame temperatures or by combustor geometry changes to limit the time at high flame temperature.

Smoke and other particulate matter is controlled by metering the air and fuel carefully in the combustor and most modern combustors produce negligible smoke.

TABLE III-20 CURRENT GAS TURBINE EMISSION LEVELS

Size 30 MW
Turbine Inlet Temperature 2000°F
Distillate Fuel - Direct Fired

Constituent	Emission level lbs/million BTU fuel
SOx	0.52
NOx	0.4
Particulates	NIL

Physical Characteristics

The physical characteristics of the gas turbine and heat recovery system are shown in the table below.

TABLE II-21

CURRENT GAS TURBINE PHYSICAL CHARACTERISTICS

Footprint	Volume	Weight
FT^2/kW	FT^3/kW	#/kW
0.05	1.57	15.

These data include the gas turbine and electric generator, housings, inlet and exhaust silencers and auxiliaries and do not account for necessary clearances, roadways, etc.

ADVANCED GAS TURBINES, LIQUID-FUELED/DIRECT-FIRED

Included with the advanced gas turbine is a fuel treatment subsystem which processes the input liquid fuels for delivery to the modified direct-fired combustor. In the case of residual petroleum fuel, the fuel treatment plant has been fairly well defined. Water is added to the residual oil to dissolve any alkali metal salts present. The water and salt is then removed by centrifugal separators or other means. The fuel is then treated by addition of a vanadium inhibitor and, perhaps, clean water and emulsifiers to promote clean combustion.

The base point advanced gas turbine chosen for this study has a 2500°F turbine inlet temperature, pressure ratio between 14:1 and 18:1, and 10 MW output. To minimize the number of parts both axial flow and centrifugal flow compressor stages are employed. By increasing the turbine inlet temperature and compression ratio relative to current gas turbines, the physical size of the advanced machines is reduced, the specific power is increased, and the conversion efficiency is improved. Figure III-61 indicates the improvements in heat rate and specific power for the advanced technologies anticipated in the 1985-2000 time period.

Realistic gas turbine component performance parameters have been assumed in the study to account for the compressor air bleed used to cool the turbine vanes and blades. Development of water-cooled turbines is underway at the present time and could be available in the 1985-2000 time period. If water cooling becomes available, the gas turbine performance may be higher than that used in this study because the performance decrement associated with water cooling is much less than it is for air cooling.

Performance Characteristics

The advanced gas turbine engine energy conversion system performance is shown as the fraction of fuel energy input for liquid fuels in Figure III-62. The three cases show essentially the same overall conversion efficiency with the electrical power portion improving slightly with increasing pressure ratio. This is due to the higher work extraction in the turbine as indicated by the decreasing turbine discharge temperature.

The higher electrical efficiency and higher overall fuel utilization of the advanced gas turbines compared to current gas turbines is due to their higher specific power. With higher specific power and a given level of stack temperature, the stack loss is reduced, improving the overall fuel utilization.

Figure III-63 shows the off-design performance variation of the advanced 18:1, 2500°F, engine in terms of heat rate and power output level. The characteristic increase in heat rate with power reduction is evident.

Estimated Costs

The capital costs for the advanced gas turbine systems fired by petroleum and coal-derived boiler fuel are shown in Figure III-64. This chart shows the dollar per kilowatt cost for three different pressure ratios. Electrical generator costs were based on Figure III-59 while heat recovery boiler costs are shown in Figure III-60. The advanced gas turbine and installation costs are shown in Figure III-65. The basic advanced engine cost includes a 10% factor for heavy fuel treatment based on the current cost of fuel treatment and the characteristics of the advanced gas turbine. This factor was applied also to the coal-derived oil case.

The cost of the advanced gas turbine itself was estimated from the current cost and specific power to the advanced rating by increasing the cost for high temperature hardware and then adjusting to account for the increased specific power. In

addition, the cost was raised by a 3% factor for a NOx control combustor. There is little variation in cost from one design option to another so a single set of costs was used in the computer analysis.

The estimated operating and maintenance costs for the two cases are tabulated in Table III-22 below.

TABLE III-22 ADVANCED GAS TURBINE OPERATING AND MAINTENANCE COSTS

Fuel Type	Combustion System	O & M Cost Mils/kWh
Petroleum Residual	Direct	2.8
Coal-Derived Boiler	Direct	2.8

These rates were estimated by reducing the 2.5 mils/kWhr for current gas turbine to 2.0 mils/kWhr for advanced engines and adding 0.8 mils/kWhr for utilizing the heavy fuels and operating costs of the treatment plant.

Emissions

The estimated emission levels for advanced gas turbines with boiler-type fuels are presented in Table III-23. The SOx emissions represent the additional sulfur assumed in the coal derived fuel. The NOx level assumes that effective measures of NOx control are developed by 1985-2000, especially in the case of coal-derived fuel where the fuel bound nitrogen content is high. Several combustor concepts are under study to achieve low conversion rates of fuel bound nitrogen into NOx in the combustion process. Basic to most concepts is a rich-burn phase where insufficient oxygen is present to allow NOx formation. This is followed by a rapid combustion phase where the rest of the fuel is burned by admitting additional air

(lean burn) and insufficient time at high temperature limits the thermal NOx produced. The particulate matter in the fuel was assumed to pass through the combustor. It may be possible, especially in the coal-derived liquid case to reduce the particulate level by additional filtering in the fuel treatment plant.

TABLE III-23 ADVANCED GAS TURBINE EMISSION LEVELS
lbs/million BTU Fuel

Fuel	SOx	NOx	Particulates
Petroleum Residual	0.76	0.5	0.03
Coal-Derived Boiler	0.82	0.5	0.10

Physical Characteristics

The physical characteristics of the gas turbine, generator, heat recovery system, housings and silencers are summarized in Table III-24. The advanced gas turbines are generally smaller than the current gas turbines due to the increased specific power.

TABLE III-24 ADVANCED GAS TURBINE PHYSICAL CHARACTERISTICS

Fuel Type	Footprint FT^2/kW	Volume FT^3/kW	Weight #/kW
Petroleum Residual	0.030	1.05	10
Coal Derived Boiler	0.030	1.05	10

Cogeneration Applicability

The direct-fired gas turbine is very adaptable to industrial cogeneration applications and has been used in this way in process industries. The high temperature exhaust gases, about 1,000°F, can raise high temperature steam or, in some applications, can be used directly in the industrial process. Since only one quarter of the oxygen in the air is consumed in the gas turbine, the exhaust gases can be used as preheated air for industrial furnaces or for raising larger amounts of steam. The total fuel utilization (electricity and useful heat) without supplemental firing is about 80 percent and is limited by the thermal losses in the stack. The advanced technology with higher turbine inlet temperatures and higher specific power reduces the magnitude of the stack losses.

The advanced gas turbine can consume boiler-grade fuels, either coal-derived or petroleum based, with emission levels consistent with the specifications.

The gas turbine-generator can be grid connected or operated independently. It offers operational flexibility and can respond to load changes rapidly. At part power, the specific power is reduced and the turbine exhaust temperature may be 100°F lower reducing the overall fuel utilization. Since the gas turbine is relatively small and light, maintenance options are enhanced. A powerplant can be changed easily and quickly. Since the gas turbine is a factory-assembled, modular unit, additions to meet expanding requirements are straightforward.

Future Developments

Gas turbines have demonstrated the basic performance, endurance and economic characteristics for successful cogeneration operation. However, to improve fuel flexibility and to extend cogeneration applicability to a wide range of industrial situations, the following technical developments should be pursued:

1. Development of materials and/or cooling systems to provide up to 2,500°F turbine inlet temperature with the desired durability. Advanced air cooling or water cooling systems could provide this capability.

2. Development of rich-lean staged combustion systems to control nitrogen oxide emissions within acceptable levels for both petroleum-based and coal-derived boiler-type fuels.

3. Development of combustion and turbine systems to operate with heavy boiler-type fuels for the desired life.

4. Development of smaller components while maintaining high component efficiency levels.

ADVANCED GAS TURBINE - GASIFIED COAL FUEL

Conversion System Description

There are a variety of methods whereby coal could serve as gas turbine fuel. Since gas turbines have operated with gaseous fuels for many years, one means of using coal is to first convert the coal to gas and then consume the gaseous fuel in the gas turbine. Such a system was included in the study. The gas turbine engine used the same technology as the advanced direct-fired, liquid fueled, gas turbines (energy conversion systems 12 and 13). The gasifier is an entrained flow, air-blown unit which converts coal into a stream of essentially hydrogen, carbon oxides, and nitrogen. A schematic diagram of the system is presented in Figure III-66.

Air is bled from the compressor, cooled and compressed to a pressure about 150 to 200 psi above compressor discharge -- the operating pressure of the gasifier. The high temperature (2400°F) gasifier discharge is cooled by raising steam for the industrial process. Ash and soot are removed in a water wash. The cooled gasifier stream passes through a desulfurizer before being injected into the gas turbine main combustion chamber. A turbine in the fuel line upstream of the combustor (not known in Figure III-66) can be used to drive the bleed air compressor.

The design point the advanced gas turbines with gasified coal used 2500°F turbine inlet temperature and 18:1 pressure ratio. The power output level of the gas turbine is nominally 100 MW. The turbine is assumed to have advanced air cooling. As discussed in the previous section, water-cooling techniques are being pursued and if available would result in improved performance.

An air-blown gasifier and gas turbine cycle with a 2400°F turbine inlet temperature and a 17:1 pressure ratio were chosen from the literature to serve as a base point. These values (and effects on cost) were modified to apply to the base values of 2500°F turbine inlet temperature and 18:1 pressure ratio for this study. Both values are reported where applicable below.

Other gasifier systems could have been chosen and results would vary somewhat. For example, a medium BTU gas could be made by using an oxygen-blown system. The resulting gas has a higher flame temperature and is a more difficult NOx emission problem but could have advantages for cogeneration. A central gasifier could serve several cogeneration sites because it is economical to transfer medium BTU gas up to several miles. This option, however, increases complexity when evaluating various cogeneration energy conversion systems and was not used in this study.

<u>Performance Characteristics</u>

The advanced gas turbine, integrated gasifier performance is shown in Figure III-67. The electrical efficiency of 18 - 20% is much lower than that of the previous simple cycle gas turbines. This is because of the additional heat from the water and/or steam used to cool the exothermic gasification process. Cooling of the gasifier produces an additional component to the process heat supplied by the turbine exhaust. In combined cycle systems, for which the integrated gasifier cycles are usually studied, the electrical efficiency is much greater (of course, then there is no thermal energy available for the process). Combined cycles are examined in later sections of this report.

The total fuel utilization is close to, but slightly less than, the simple cycle liquid fueled engines. This is primarily due to additional unrecoverable losses from the gasifier and cleanup systems and a penalty for adding steam to the gas stream which results in a higher latent heat loss.

The electrical output of the 18:1 pressure engine is slightly higher than that of the 17:1 pressure ratio case found in the literature, Figure III-67. The off design performance would be similar to that shown previously in Figure III-63, but would be complicated by the off design operation of the gasifier.

Estimated Cost

The cost of the gas turbine engine is shown in Figure III-68. The cost includes modifications to the engine to bleed off air to the gasifier and for modified combustors to burn low BTU gas.

The gasifier cost for various gas turbine power outputs is shown in Figure III-69. The cost of these gasifiers was obtained from the literature where possible and scaled to provide the necessary power range. The heat recovery boilers were based on the heat recovered from the gas turbine exhaust stream with additional process heat being obtained from the gasifier.

The total cost and breakdown of the gas turbine and gasifier costs are shown for the two systems studied in Figure III-70. The assumed cost of the gasifier and the cleanup system is much greater than the cost of the gas turbine system. This indicates that it is most economical to operate these systems on a continuous load basis to spread the gasifier cost over as many hours as possible.

The operating and maintenance cost of the gasified coal system is presented in Table III-25.

TABLE III-25 - COAL GASIFIER ADVANCED GAS TURBINE
OPERATING AND MAINTENANCE COST

Fuel Type	Combustion System	O&M Cost Mils/Hr.
Coal-Gasified	Direct	3.0

The operating and maintenance rate reflects a maintenance rate of 2.0 mils/kWh on the gas turbine but additional costs for the gasifier.

Emissions

The emissions characteristics of the advanced gas turbine coal gasification system are shown in Table III-26 below.

TABLE III-26 - ADVANCED GAS TURBINE-COAL GASIFICATIONS
lbs/million BTU

Fuel Type	SOx	NOx	Particulates
Coal-Gasified	0.82	0.5	Nil

The SOx emissions represent primarily the emissions from a waste stream in the gasifier. The sulfur level in the gas burned in the gas turbine is very low with low SOx produced. The NOx produced is primarily thermal NOx and is minimized by the fact that flame temperatures in the low BTU gas are very low.

Physical Characteristics

The physical characteristics of the gas turbine for the gasified coal system are shown in Table III-27 below.

TABLE III-27 - ADVANCED GAS TURBINE - COAL GASIFIER PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/kW	Volume FT^3/kW	Weight #/kW
Coal-Gasified	0.03	1.05	10

These physical characteristics apply to the major components of the gas turbine equipment and do not include clearance spaces, roads or area required by the gasification system.

Cogeneration Applicability

The gasified coal direct-fired gas turbine is potentially very adaptable to industrial cogeneration applications. In addition to the high temperature exhaust gases, about 1000°F, there is an additional source of high temperature (2000-2400°F) heat from the fuel gas leaving the gasifier. A large number of configurations using various combinations of heat sources and fuel sources is possible to meet process requirements. Coal gasification has been assumed for the CTAS study, but alternate carbonaceous fuels such as wood or grain can be gasified.

The gasifier, depending on its type and whether it is air or oxygen blown, can use different types of coal to produce clean low or medium BTU fuel gas. Emission levels can be kept low, especially with the low BTU gas assumed in the CTAS study because of its low flame temperature. Coal ash and sulfur removed from the

coal must be disposed of, however. A central, medium BTU gasifier could supply gas for a number of different cogeneration systems within several miles.

The start and warm up times of large gasifiers is relatively slow, but once operation conditions are reached, operational flexibility and load response is good. The system can be grid connected or operated independently and the gasifier can be designed to produce gas in excess of that required by the gas turbine for power to meet additional thermal needs. Gasifiers under development are generally of large size, greater than 10 MW for economy of scale.

Future Development

Experimental development of gasifier/gas turbine systems for utility applications is currently being supported by government and private funding. Additional development of smaller industrial gasifiers is also underway, but at lower funding levels. To provide for a wider range of cogeneration applications, the following technical developments should be pursued:

1. Full scale operation of an integrated gasifier/gas turbine system.

2. Development of combustion systems using low BTU gas at turbine temperatures up to 2500°F. These systems are expected to have very low NOx emissions.

3. Development of gasifiers using different types of coal, wood, very heavy petroleum and byproduct fuels.

4. Development of both large and small gasifiers to permit a wide range of cogeneration applications.

ADVANCED GAS TURBINE, DIRECT COAL-FIRED, PRESSURIZED FLUIDIZED BED

System Description

The advanced gas turbines which operate on coal directly fired in a pressurized fluidized bed (PFB) assumed for this study are similar in component performance technology to the other advanced gas turbines used in this report. The turbine inlet temperature, however, is much lower - 1600°F. This results in a lower specific power and lower electrical efficiency.

A split flow PFB was assumed as shown in Figure III-50. In this system part of the compressor air flows into the pressurized fluidized bed for combustion with coal in the pressence of dolomite. The other part of the compressor air passes through tubes in the bed where it is heated. The combustion air is mechanically cleaned and mixed with the uncontaminated air and passed through the turbine. Process heat can be obtained from the exhaust heat of the turbine and from air or steam tubes immersed in the fluidized bed.

The base point for this study was assumed to be 1600°F turbine inlet temperature and 10:1 pressure ratio. The nominal engine size was the 60 MW. This is, of course, physically a large engine capable of over 100 MW at normal turbine inlet temperatures.

Performance Characteristics

The advanced gas turbine pressurized fluidized bed performance is shown in Figure III-71.

In design options 1, 2, and 3, the electrical efficiency varies between 22 and 26% as the pressure ratio increased from 6 to 10. The overall fuel utilization is between 73 and 76% over the same range. This is slightly lower than the fuel utilization for the higher turbine temperature gas turbines due to higher heat losses

from the PFB and its waste streams. While the electrical efficiency of the higher pressure ratio gas turbines is greater than the lower pressure ratios, the exhaust temperature is lower resulting in lower quality process heat.

Also included (as design option 4) is the case where a portion of the process heat is provided in the high temperature fluidized bed. Fuel utilization of 78% is obtained with an electrical efficiency of about 19.5%.

Off-design performance of the advanced gas turbines used in this study is shown in Figure III-72.

Estimated Cost

The cost of the gas turbine is shown in Figure III-73. The cost is adjusted from current engine levels by reducing the cost of the turbine and scaling according to the specific power output. The difference in cost between 1500 and 1600°F turbine inlet temperature is small, so the same cost is used for the PFB and AFB coal-fired gas turbine systems.

The exhaust heat recovery system cost was obtained from Figure III-60. The total PFB gas turbine systems costs are shown in Figure III-74. The coal fluid bed heat source and hot gas cleanup systems were not included in the gas turbine account, but are included in Volume IV.

The estimated operating and maintenance cost of 3 mils/kWh reflects a very low rate for the basic gas turbine because of its simple design operating at low temperatures. This rate, however, is approximately doubled by the relatively low specific power, the potentially corrosive atmosphere, and the effect of particulate erosion.

Emissions

The estimated emissions for the advanced direct coal-fired gas turbines system are included In Table III-28.

TABLE III-28 - ADVANCED GAS TURBINE - PRESSURIZED FLUIDIZED BED EMISSIONS LBS/MILLION BTU

Fuel Type	SOx	NOx	Particulates
Coal	1.20	0.2	0.001

The SOx emissions were calculated by assuming an 85% sulfur removal in the bed for the specified coal. The NOx emissions require that only a fraction of the nitrogen in the coal is converted to NOx. This fraction has been shown to be generally obtainable in fluidized bed testing. The particulate emissions shown are within the specfied limits. A value near this limit is necessary in order to obtain satisfactory turbine life, at least with particulates in the one micron range. Smaller particles can be removed from the gas turbine exhaust in a bag house or electrostatic precipitator.

Physical Characteristics

The physical properties of the direct coal fired gas turbines without the heat source are presented in Table III-29.

TABLE III-29 - ADVANCED GAS TURBINE - PRESSURIZED FLUIDIZED BED PHYSICAL CHARACTERISTICS

Fuel Type	Footprint FT^2/KW	Volume FT^3/KW	Weight #/KW
Coal	0.07	2.28	21

Cogeneration Applicability

The pressurized fluidized bed (PFB) combustor has many advantages over competing systems for cogeneration applications. When fired with coal, dolomite is also injected into the bed where the calcium in the dolomite reacts with most of the sulfur in the coal removing the sulfur as a solid preventing SOx emissions. Turbine inlet temperatures are limited to 1500-1700°F (1600°F was used in the study) to maximize sulfur capture by the dolomite. At these temperatures, the chemical reactions of nitrogen in the coal favors formation of N_2, keeping production of NOx to acceptable levels. Also at these temperatures, the coal ash is a dry solid, well below slagging temperatures, permitting its removal by cyclones and precipitators.

Two sources of high temperature heat are available for processes, the bed itself at 1500-1700° and the gas turbine exhaust at 800-1000°. The PFB is physically smaller than an AFB and units can be added as modules to provide size flexibility. In some cases the coal feed system should be sized for the final projected capacity to obtain the economies of scale. The PFB can also combust a wide variety of fuels.

The start and warm up are slow, but when operating on a grid, or isolated, the response to load changes is rapid. Sudden shutdowns such as drop load can be accommodated since the coal in the bed amounts to about 1% of the bed mass and quickly burns out without temperature overshoot.

Future Development

Pressurized fluidized bed (PFB) combustors are under active development. A pilot plant is being constructed and numerous other studies are also under way on various PFB components and commercial plant utility designs. To permit availability of PFB systems for cogeneration applications, the following technical developments should be pursued:

1. Full scale operation of PFB/gas turbine systems.

2. Development of pressurized coal/dolomite feed systems.

3. Experimental studies of corrosion and erosion of high temperature heat transfer tubes and internals of the PFB in the combustion region.

4. Development of high temperature particulate cleanup systems.

5. Experimental studies of turbine corrosion, erosion and deposition due to the PFB combustion products.

ADVANCED GAS TURBINE, INDIRECT COAL FIRED - ATMOSPHERIC FLUIDIZED BED

Conversion System Description

Advanced gas turbines can also be operated on coal fired in an atmospheric fluidized bed. Compressor air is directed through tubes immersed in the bed, heated to approximately 1500°F and expanded through the turbine. The exhaust air, which is free of the products of combustion, can be used directly in a process or used to heat steam or some other fluid. The lower temperature of 1500° assumed for the atmospheric fluidized bed relative to the pressurized fluidized bed represents the differing heat source characteristics, (see Volume IV).

Two systems shown in Figure III-50 were considered in this study. One uses a simple air heater independent of the gas turbine. The other semi-enclosed system uses all or a portion of the turbine exhaust air in the AFB to provide the combustion air. This avoids separates air heaters and blowers.

The base point for this study assumed a 1500°F turbine inlet temperature and a pressure ratio of 10:1. Like the direct coal-fired PFB gas turbine, the nominal output about 60MW.

Performance Characteristics

The estimated performance of the advanced gas turbine design options with an atmospheric fluidized bed coal combustion heat source is presented in Figure III-75.

These options employ the semiclosed system in Figure III-50 and have varying amounts of exhaust gas used in the combustion process. For design option 3, all of the exhaust gas is used as combustion air resulting in the maximum process steam for a given electrical output. The fuel utilization efficiency is 81.5% while the electrical efficiency is 11.3%.

Reducing the combustion air by 1/3 and 1/2 respectively results in design options 2 and 1. It is assumed that the exhaust air that is not bypassed can be used down to any temperature desired since there is no contamination of this air. Assuming a 60°F utilization of this air, an overall fuel utilization of 83% can be obtained.

Estimated Cost

The cost of the gas turbine engine is shown in Figure III-73. Exhaust heat recovery costs are included in Figure III-60. The estimated costs of the advanced gas turbine based on a coal-fired atmospheric fluidized bed heat sources are presented in Figure III-76. The cost of the heat sources is not included in Figure III-76 but is presented in Volume IV.

The estimated operating and maintenance cost of, 1 mil/kWh reflects the very low operating and maintenance cost to be expected for a simple advanced gas turbine operating on clean hot air with a turbine inlet temperature of 1500°F.

Emissions

The only source of emissions is the heat source and these data are presented in Volume IV, Table IV-41.

Physical Characteristics

The physical characteristics of the advanced gas turbine using a coal-fired atmospheric fluidized bed heat source are presented in Table III-30 without the heat source. The physical description of the heat source is included in Volume IV.

TABLE III-30 - ADVANCED GAS TURBINE - ATMOSPHERIC FLUIDIZED BED PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal	0.07	2.28	21

Cogeneration Applicability

The atmospheric fluidized bed (AFB) combustor can heat compressed air supplied from a gas turbine; the heated compressed air is then expanded through the turbine. Since the air is heated in tubes within the bed, it is uncontaminated and the high temperature exhaust can be used in processes requiring hot air which is free of combustion products. The AFB itself can also be used as a high temperature heat source since it operates at about 1500°F. This temperature, somewhat lower than the PFB, maximizes sulfur capture by the limestone which is fed to the bed. The calcium in the limestone forms calcium sulfate with most of the sulfur contained in the coal and is removed as a solid with the ash. NOx emissions are low because bed temperatures favor formation of N_2 from the nitrogen contained in the coal. Flyash and particulates in the combustor effluent can be removed by conventional means.

The AFB is large when compared to a conventional gas turbine combustor or a PFB. It should, however, be smaller than a conventional coal fired boiler. In most coal fired systems, the land area needed is predominantly sized by the coal

pile and handling systems, so the size disadvantage of the AFB is not great. The AFB can combust a wider variety of fuels than the PFB since the fuel does not have to be injected into a high pressure region.

The start and warm up time of the AFB is fairly long. Combustion air is supplied by a blower or by using a portion of the exhaust from the gas turbine. Response to load changes is good and the AFB can be operated on a grid or isolated. Drop loads can be accommodated without temperature overshoot problems since the coal makes up only about 1% of the mass of the fluidized bed.

Future Development

The commercialization of atmospheric fluidized bed (AFB) combustors is high on the Department of Energy priority list. Both utility and industrial development programs are underway, backed by several component and study programs. The technical development programs that need to be pursued are generally either in the scaling up or refinement categories. These programs are:

1. Successful testing and accumulation of operating hours on the current demonstration units.

2. Development of fuel injection systems to increase the bed area/injector.

3. Experimental studies of high temperature corrosion and erosion in the fluidized bed combustor.

4. Experimental verification of design parameters leading to the commercial availability of AFB's for cogeneration applications.

CLOSED CYCLE GAS TURBINE, INDIRECT COAL FIRED ATMOSPHERIC FLUIDIZED BED

Conversion System Description

The closed cycle gas turbine, or Brayton cycle, is an interesting candidate for cogeneration applications because heat recovery cools the exhaust gas and further cooling up stream of the compressor is minimim. The closed cycle parameters are not limited by ambient conditions. Two closed cycle gas turbine heat sources were included in the study: (1) coal-fired atmospheric fluidized bed with a turbine inlet temperature of 1500°F and (2) liquid boiler fuel furnace with 2200°F turbine inlet temperature. This section describes the closed gas turbine with the lower temperature, coal-fired heat source (energy conversion system number 18).

Since the cycle is closed, any gas could serve as working fluid. A heavy molecular weight gas, such as argon, reduces the size of the turbo-machinery at the expense of the heat transfer components. A light gas, such as helium, minimizes the size of heat transfer components but requires more extensive turbo-machinery. Air has the advantage of reducing sealing requirements and mechanical complications. For this study air and helium were considered.

The size of the turbo-machinery is also dependent upon the pressure level. A maximum pressure of 600 psi was selected.

The conventional open cycle gas turbine is limited to air as working fluid and is limited by the ambient pressure and temperature. The closed cycle permits a choice of compressor inlet temperature and pressure (or pressure ratio). Pressure ratios of 3:1 and 6:1 were considered and compressor inlet temperatures of 190 and 300°F were included.

With low pressure ratios (particularly with helium as the working fluid) regenerators or recuperators can improve traditional cycle efficiency. Figure III-77 presents simple schematic diagrams of the closed cycle with and without a regenerator.

To provide guidance in the closed cycle gas turbine design selections, parametic design studies were conducted by United Technologies Research Center and Power Systems Division using the component performance listed in Table III-31.

Table III-31

CLOSED BRAYTON CYCLE COMPONENT PERFORMANCE PARAMETERS

	Air	Helium
Compressor Efficiency, %	88	90
Turbine Efficiency, %	91	90
Generator Efficiency, %	96	96
Regenerator Effectiveness, %	85	85

The results of these parametric studies are depicted in Figures III-78, III-79, and III-80. Figure III-78 indicates that the electrical output increases as the pressure ratio increases but the quality of the recovered heat is reduced. The case with a regenerator is included in Figure III-79. The regenerator increases the electrical output but reduces the temperature entering the heat recovery heat exchanger and lowers the quality of the recovered heat. No 700°F steam is available at any pressure ratio studied. Raising the compressor inlet temperature, Figure III-30, increases the high temperature heat recovered at a modest penalty in electrical output.

Performance Characteristics

Five closed Brayton cycle designs were selected to cover a range of possible industrial applications. Two design options used helium as the working fluid and a pressure ratio of 3:1. One of these employed a regenerator. Three options used air as the working fluid, a pressure ratio of 6:1, and one of these also used a regenerator. The compressor inlet temperature was 190°F except for one case at 300°F. The performance of the three design options without the regenerator is presented in Figure III-81. The two cases with the regenerator are included in Figure III-82. Design option 2 with the high compressor inlet temperature provides the largest amount of high temperature steam and the highest overall fuel utilization - 84%. The regeneration cases provide the highest electrical output.

Estimated Costs

The estimated costs of the closed cycle gas turbines operating with air were based upon the costs of conventional gas turbines, Figure III-59, accounting for variations in turbine inlet temperature, pressure, and compressor inlet temperature, size, and working fluid. The specific power (electric output divided by air mass flow) varies with turbine inlet temperature and cycle pressure ratio as indicated in Figure III-61. The 1500°F turbine inlet temperature increases the cost by 30 percent over the cost of a current technology gas turbine with a turbine inlet temperature of 2000°F. Since the turbo-machinery is basically a volume flow device, the operating pressure has a direct effect on turbo-machinery size for a given output. With 600 psi turbine inlet pressure and 100 psi compressor inlet pressure in the closed cycle compared with 200 and 15 psi respectively in the conventional case, the closed cycle turbo-machinery is 1/3 to 1/6 the size of the current technology. The higher compressor inlet temperature in the closed cycle requires a 10 to 20 percent increase in size. Smaller size turbo-machinery has higher specific cost, as illustrated in Figure III-59, and the closed cycle cost estimates included this effect. The generator cost estimate was based on Figure III-59 and the heat recovery heat exchanger estimate is based on Figure III-60.

c-2

To establish the regenerator, or recuperator, cost estimate a computer aided shell and tube type heat exchanger design was established based upon material structural properties and average fluid characteristics. Cost estimates for closed cycle gas turbines using air as the working fluid are presented in Figure III-83. The system with the regenerator costs about 8 percent more than the configuration without this added element.

The turbo-machinery cost estimate for the helium working fluid cases was based upon the revised volume flow in light of the change in molecular weight and other fluid properties (such as specific heat, specific heat ratio, etc.). As a practical matter, the design pressure ratio was reduced to 3:1 with helium so that typically the helium turbo-machinery specific cost was about 2 to 3 times the cost of the equipment used with air. The regenerator was typically about half the cost with helium as with air. The net estimated cost effect of using helium instead of air in the closed cycle gas turbine is small - within a few percent.

Since the cost estimates for the closed cycle gas turbine design options indicated small cost variations, a single set of cost estimates was used in the computer analysis. The coal-fired atmospheric fluidized bed heat source cost estimate is not included in Figure III-83 and is presented in Volume IV.

The estimated operating and maintenance cost of 0.8 mil/kWh reflects the very low cost to be expected for a closed cycle gas turbine operating at a moderate temperature on clean air.

Emissions

The only source of emissions is the heat sources and these data are included in Volume IV, Table IV-41.

Physical Characteristics

The physical characteristics of the closed cycle gas turbine are included in Table III-32. The physical properties of the heat source are presented in Volume IV.

TABLE III-32 - CLOSED CYCLE GAS TURBINE - ATMOSPHERIC FLUIDIZED BED PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal	0.03	1.05	10

<u>Cogeneration Applicability</u>

The closed cycle using air or helium as a working fluid and a high temperature gas to gas heat exchanger can provide several sources of high temperature heat to match a wide variety of cogeneration applications. In the CTAS results the air and helium closed cycles were evaluated approximately equal so emphasis of this section is on the air cycle. In a closed cycle, the continuously circulating working fluid must be cooled from the turbine exhaust temperature to the compressor inlet temperature. All or any portion of this heat is available for process heat. The closed cycle itself has no emissions; all emissions are due to the furnace. Fuel flexibility is also determined by the furnace design and is not limited by the turbine.

Startup time is relatively slow but the response to load changes is good as is characteristic of gas turbine systems. Part power efficiency can nearly equal full load efficiency. For maximum efficiency at part power the system operating pressure can be lowered giving a nearly flat efficiency versus percent power characteristic.

The closed cycle gas turbine is physically smaller than open cycle gas turbines because of the high system pressures. The heat exchanger equipment and connecting pipes more than overcome this size advantage resulting in a system that has a lower power density than an open cycle gas turbine. The small physical

size of the gas turbine for a given power level limits the small power applicability of the closed cycle gas turbine system. For a helium cycle, the gas turbine would be physically even smaller and probably less expensive, but the heat exchangers were found to be more expensive. Also, the aerodynamic design of the helium turbomachinery would be completely different than the air turbomachinery.

Future Developments

The closed cycle gas turbine using air could be developed for cogeneration applications with low technical risk. The moderate turbine inlet temperature of 1500° requires no turbine cooling air and the overall pressure ratio is low. The main development areas are in the metallic air heat exchanger which has the same technology as the AFB indirect fired system. The closed cycle helium design would introduce a moderate amount of technical risk. While of a minor nature, the following gas turbine technical developments should be pursued:

1. Development of bearings, lubrication and sealing systems for the higher than normal pressures and temperatures.

2. Analysis of performance and size characteristics to meet the projected cogeneration applications.

CLOSED CYCLE GAS TURBINE - INDIRECT FIRED - HOT GAS FURNACE

Conversion System Description

The closed cycle gas turbine with the hot gas furnace (energy conversion system number 17) is the same in principle as the previous lower temperature case (number 18) with the exception of the heat sources. The energy used to heat the circulating working fluid is derived from a hot gas furnace that is fired with petroleum or coal-derived boiler fuel. This furnace, with an advanced ceramic heat exchanger, provides a gas turbine inlet temperature of 2200°F. Figure III-84 is a schematic representation of this arrangement.

Performance Characteristics

Five closed gas turbine design options were developed to provide for a range of industrial applications. These options use air as the working fluid and operated at pressure ratios of 6:1 and 14:1. A regenerator was included in one option. Two designs used helium as the working fluid, one with a regenerator, with pressure ratios of 4:1 and 6:1. In all cases the turbine cooling flow was 15 percent. The components performance parameters are provided in Table III-31.

The performance of the non-regenerative and regenerative cases is shown in Figures III-85 and III-86, respectively. The performance with the hot gas furnace systems is higher than for comparable cases with the coal-fired atmospheric fluidized beds because of higher turbine inlet temperatures.

Estimated Costs

The estimated costs for the closed cycle gas turbines with the high temperature heat sources were developed as described in the previous section for the lower temperature systems. The estimates using air as the working fluid, Figure III-87, indicate that the regenerator can have an adverse effect on costs. As in the low temperature case, the use of helium instead of air does not affect the estimated costs appreciably. Since the cost estimates of the high temperature closed gas turbine design options indicated modest variations, a single set of cost estimates was used in the computer analysis. The high temperature heat source cost estimates are not included in Figure III-87 but are reported in Volume IV.

The estimated operating and maintenance cost of 1.2 mil/kWh reflects the very low cost to be expected for a closed cycle gas turbine operating at a moderate temperature on clean air.

Emissions

The only source of emissions is the heat source and this emission data is presented in Volume IV, Table IV-25.

Physical Characteristics

The physical characteristics of the closed cycle gas turbine are presented in Table III-33. The physical characteristics of the heat source are presented in Volume IV.

TABLE III-33 - CLOSED CYCLE GAS TURBINE - INDIRECT FIRED - HOT GAS FURNACE - PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Petroleum Residual or Coal-Derived Boiler Fuel	0.03	1.05	10

Cogeneration Applicability

The closed cycle using air or helium as a working fluid and a high temperature ceramic gas to heat exchanger is generally applicable to a wide range of cogeneration applications. The air and helium closed cycles were evaluated to be about equal in the CTAS study and the emphasis in this section is on the air cycle. The continuously circulating fluid in the closed cycle is exhausted from the turbine at a high temperature and all or any portion of this heat can be used for processes before the flow is cooled to the compressor inlet temperature. The emissions are due to the furnace which is assumed to be liquid fired to 2200°F.

Startup time is relatively fast because it is liquid fired and the response is good when grid connected or operating isolated. Part load efficiency is practically equal to full load since the system pressure level can be changed while holding other cycle parameter constant. The efficiency as a function of percent power is essentially constant.

The closed cycle turbomachinery is physically smaller than open cycle turbomachinery for a given power output because of the elevated system pressure. This advantage is nullified by the added piping and heat exchangers required. The small physical size of the turbomachinery limits the minimum economic power of the closed cycle. Using helium as a working fluid would result in even small physical sizes for a given power.

Future Development

The closed cycle gas turbine using air at a 2200°F turbine inlet temperature would constitute a technical advance over current industrial gas turbines. Turbine cooling would be required to reduce metal temperatures to obtain the necessary creep life. The recirculating gas would be free of products of combustion and atmospheric contaminants, but oxidation could still determine permissible surface metal temperatures. Nitrogen could be used as a working fluid to avoid oxidation. The ceramic heat exchanger and ducting to handle 2200°F gas are the major development areas. The following gas turbine technical developments should be pursued:

1. Experimental studies and designs to permit 2200°F turbine inlet temperatures.

2. Investigation of oxidation of materials for the high temperature closed cycles.

3. Development of bearings, lubrication and sealing systems for the higher than normal temperature and pressures.

STEAM INJECTION CYCLES - ADVANCED GAS TURBINE - DIRECT FIRED PETROLEUM OR COAL DERIVED LIQUID

Conversion System Description

In a steam injection cycle, steam is produced by the exhaust heat of the gas turbine, is injected at high pressure into the gas turbine, and expands through the turbine to produce additional power. In this direct fired advanced gas turbine design the steam raised by the exhaust is injected into the compressor exit air at the compressor exit temperature. The steam, of course, would be throttled through a control valve to match desired flows and pressures.

Air to steam ratios of 20:1 and 10:1 were selected as representative for cogeneration systems. These values would augment the simple cycle gas turbine power considerably but would leave additional steam available for the process. The injected steam is superheated to 2500°F in the combustor and provides considerable augmentation. The increased turbine mass flow also provides additional exhaust heat which raises the potential steam production from the exhaust.

The cost of the steam injection cycle is potentially lower than a combined cycle because the steam turbine is eliminated. The gas turbine, of course, must be modified to accept the steam and enlarged to produce the increased power. The performance is higher than that of the simple cycle, but lower than a combined cycle.

Steam injection is used to control NOx in gas turbines burining heavy fuel. The air to steam ratios are generally about 50.

Performance Characteristics

Calculating the performance of steam injection cycles involves additional parameters beyond those used for simple cycles and steam bottoming combined cycles. For this direct fired system design saturated steam was raised to the temperature of

the compressor exit flow. This allows a high enough pressure to inject the steam into the compressor flow. Enough fuel is then burned in the steam and the compressor air (minus that bypassed for turbine cooling) to raise the mixture to the desired turbine inlet temperature. Using gas properties for the mixed steam and air, the flow is expanded through the turbine producing power and a high temperature exhaust stream. The heat of the exhaust is then used to raise the injected and process steam before being exhausted to the atmosphere.

The heat rate and specific power are shown in Figure III-88 for various pressure ratios and air to steam flow ratios. Two design options were selected at a pressure ratio of 18:1, one with air to steam flow ratio of 10 and the other, 20. The fuel utilization for these steam injected cycles is shown in Figure III-89.

Cost

The cost of steam injected cycles had to be based on conventional gas turbine cost characteristics because no commercial units are in production. The cost of a conventional gas turbine was determined for each major component (e.g., compressor, combustor, turbine). The components affected by the steam injection were scaled according to the volume flow of the steam and air.

The new component costs were then added up, including the estimated cost for providing for the steam injection. The result is a lower estimated specific cost for the steam injected engine. The estimated heat recovery boiler and gas turbine costs are presented in Figure III-90.

The estimated operating and maintenance cost for the steam injected gas turbine is 2.6 mils/kWh. In accordance with the study ground rules, the cost of water was not included in the estimated O&M cost.

Emissions

Although steam injection is one method of reducing NOx emissions, the conservative assumption was made that the emissions for steam injected engines are the same as the emissions for the advanced gas turbine direct-fired sample and combined cycle engines.

There is a possible environmental consideration, particularily in large installations, concerning the amount of water vapor discharged to the atmosphere.

Physical Characteristics

The physical properties of the gas turbine, generator, boiler, and housings are summarized in Table III-34.

TABLE III-34 - ADVANCED GAS TURBINE - DIRECT STEAM INJECTION - PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Petroleum Residual or Coal Derived Boiler Fuel	0.03	1.14	10

Cogeneration Applicability

Steam injection into the high pressure compressor air prior to combustion in a liquid fueled, direct fired gas turbine can provide increased electrical power while recovering thermal heat from the gas turbine exhaust. The thermal to electrical ratio is reduced relative to a simple cycle engine. The steam turbine generator and condenser system of the combined cycle can be eliminated, potentially making the steam injection cycle, economically attractive in smaller sizes than the combined

cycle cogeneration applications. The turbine must be enlarged to pass the added steam flow. Emissions are potentially easier to control with steam injection. Water is consumed and the latent heat is lost to the atmosphere.

The steam flow and compressor flow must be properly matched to the turbine to permit surge-free operation, but the design possibilities make the cycle applicable to a variety of cogeneration systems.

Startup of the system was not analyzed. It should be possible to develop a starting method that would reach a high percentage of design power in the 3 to 20 minutes typical of simple cycle gas turbines. As steam is produced, power could be increased to design values. Once on the grid or operating isolated, the response to load changes would be good. The system would be physically larger than for a simple cycle and require a water supply. Modular units could be easily added to a site.

Future Development

The incorporation of steam injection into a liquid fueled gas turbine is not technically difficult. Water injection has been used extensively in aircraft engines and water and steam injection have been used for NOx control in utility gas turbines. Accommodating higher steam flows, however, requires an enlargement of the turbine area. In addition to developing advanced gas turbines to attain 2500°F turbine inlet temperatures, the steam injection cycle requires:

1. Aerodynamic and matching studies of the gas turbine to accommodate the steam flow.

2. Experimental studies and verification of steam/air combustion and emission characteristics.

STEAM INJECTION - DIRECT FIRED PFB COAL FUEL

Conversion System Description

This system, like the direct fired liquid fuel steam injection system described in the previous section, uses steam generated by the gas turbine exhaust and superheated by the combustion of fuel to increase the gas turbine power. It was assumed that the steam was mixed with compressor air at compressor exit temperature as in the previous case. A split flow pressurized fluidized bed design was used as shown schematically in Figure III-50(C). The steam raised by the gas turbine exhaust was throttled and mixed with the compressor flow and highly superheated by passing through tubes immersed in the fluidized bed.

The heat released for a given gas turbine air flow for the steam injection cycle is greater than for the comparable simple cycle or combined cycle. A greater portion of compressor flow is directed to the combustor and more coal is burned.

Performance Characteristics

The design selections were similar to those made in the previous section. The steam, superheated to 1600°F, dramatically increases the turbine power output over the simple cycle. The heat available for the process, of course, is reduced as more steam is directed into the gas turbine. The performance of the direct coal-fired (PFB), steam injected gas turbine is included in Figure III-91.

Estimated Cost

The base advanced PFB coal combustion system and gas turbine costs were increased to match the requirements of the added steam flow and divided by the increased power output to obtain the specific power for the steam injection cycle. The assumptions made were the same as described in more detail in the previous section.

The estimated operating and maintenance costs are 2.8 mils/kWh for the steam injected gas turbine with pressurized fluidized bed direct coal combustion. In accordance with the study ground rules the cost of water is not included.

Emissions

The emissions are all based on pounds per million BTU's fired and are unaffected by the PFB steam injection cycle assumed. The coal and percentage of excess air were held constant in the combustor as the output was increased due to steam injection. The steam was assumed to be superheated inside of tubes immersed in the bed. It is possible that the SOx and NOx emissions could be reduced by introducing the steam directly into the combustion zone, but that was not assumed in this study. The estimated emissions for the steam injected, coal-fired gas turbine are presented in Table III-35.

TABLE III-35 STEAM INJECTION ADVANCED GAS TURBINE - PRESSURIZED FLUIDIZED BED - EMISSIONS LBS/MILLION BTU

Fuel Type	So_x	No_x	Particulates
Coal	1.2	0.2	0.001

Physical Characteristics

The physical characteristics of the steam injected, coal-fired gas turbine are presented in Table III-36 without the heat source reported in Volume IV.

TABLE III-36 - ADVANCED GAS TURBINE - STEAM INJECTION PRESSURIZED FLUIDIZED BED - PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal	0.07	2.47	21

Cogeneration Applicability

Steam injection into the high pressure compressor air prior to combustion in the PFB increases the electrical output by using heat from the gas turbine exhaust. The total power is increased and no steam turbine is needed. The thermal to electrical ratio is reduced and elimination of the steam turbine, generator and condenser systems reduce the system cost, especially for smaller size applications. The turbine of the gas turbine, however, must be enlarged to pass the added flow and much of the cost advantages are negated. Also, the steam exhausted to the atmosphere consumes high quality water and represents a heat loss. Emissions are low, the same as the other PFB systems.

There is considerable flexibility in temperature, pressure and amounts of steam that can be produced. For system computations, it was assumed that saturated to slightly superheated steam was introduced into the compressor. This could be easily changed for a given process.

Like the PFB without steam injection, the system start up is slow, response is good and the system can operate on a grid or isolated. A wide range of fuels could also be accommodated.

Future Development

The PFB steam injection cycle would follow the development of the base PFB cycle. In addition to the PFB future development, the steam injection cycle would require:

1. Aerodynamics and matching studies of the gas turbine to accommodate the steam flow.

2. System control development.

3. Experimental studies of combustion, corrosion, erosion and emissions as a function of steam flow rates.

STEAM INJECTION - INDIRECT FIRED AFB COAL FUEL

Conversion System Description

This system is similar to the PFB system described in the previous section, but the steam is mixed with all of the compressor air, and the mixture is heated in tubes immersed in the atmospheric fluidized bed. The atmospheric fluidized bed must be increased in size for a given gas turbine airflow to match the increased specific power obtained with the steam injection cycle. Also, as in the previous cases, the steam is throttled to match the compressor pressure and was assumed to match the compressor exit temperature.

Performance Characteristics

The steam is superheated to 1500°F along with the compressor air and the specific power of the gas turbine is again increased dramatically for the 20:1 and 10:1 air to steam flows assumed. The performance for these systems are shown in Figure III-92.

Estimated Cost

The cost estimates were based on conventional gas turbines modified to account for the steam injection as in the previous cases. The cost estimates for the coal-fired atmospheric fluidized bed heat source are included in Volume IV. The estimated operating and maintenance costs for this relatively low temperature steam injected gas turbine are 1.2 mils/kWh. In accordance with the groundrules, the cost of water was not included.

Emissions

Emissions are only dependent on the heat source reported in Volume IV, and are unaffected by the steam injection cycle.

Physical Characteristics

The physical properties of the steam injected gas turbines without atmospheric fluidized bed heat source are presented in Table III-37. The heat source is described in Volume IV.

TABLE III-37 - ADVANCED GAS TURBINE - STEAM INJECTION-ATMOSPHERIC FLUIDIZED BED - PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal	0.07	2.47	21

Cogeneration Applicability

Steam injection into the high pressure compressor air before it is heated in the AFB heat exchanger increased the electrical output and conserves heat from the gas turbine exhaust. The thermal to electrical ratio is reduced and the cost of the steam turbine, generator and condenser is eliminated. The turbine of the gas turbine, however, must be enlarged to pass the added steam flow increasing its cost. Water is consumed and the latent heat is lost to the atmosphere. Emissions are low, due only to the combustion source.

The steam temperatures, pressures and flow rates can be adjusted to meet a wide variety of applications. Saturated or slightly superheated steam injection was assumed in the study, but this could be tailored to meet other application requirements.

Startup of the AFB is slow, but load following response would be good. The system can be grid connected or operated independently. The same wide variety of fuels as in other AFB's can be accommodated.

Future Development

The AFB steam injection cycle could precede the PFB steam injection cycle. It could be applied to an AFB system with relatively little development. In addition to the normal AFB development, the steam injection cycle requires:

1. Aerodynamic and matching studies of the gas turbine to accommodate the steam flow.

2. System control development.

3. Experimental studies of corrosion and erosion of the high temperature heat exchanger in the presence of an air/steam mixture.

CURRENT COMBINED CYCLE - DISTILLATE FUEL

Conversion System Description

The exhaust temperature of current gas turbines, Figure III-52, is typically about 1000°F. In some installations, Figure III-47 and III-49(C), heat recovery boilers raise steam which is used to produce additional electricity. Such combined gas and steam turbine systems are defined as combined cycles.

If all of the available steam power were utilized for electrical power generation, all of the heat rejected would be at low temperature and of little use for most cogeneration applications. The cogeneration combined cycles, therefore, represent systems that provide some additional electrical power derived from steam and some thermal energy applicable to processes.

Current gas turbine performance parameters are applicable to cogeneration combined cycle applications with no special modifications. In this study, a gas turbine pressure ratio of 14:1 and a turbine inlet temperature of 2000°F were used for current technology. As with the other current gas turbines, distillate fuel or

natural gas is normally used. The gas turbines exhaust heat is used to raise steam at 800°F, 800 psi to drive the steam turbine.

Performance Characteristics

The performance for a representative cogeneration current gas turbine combined cycle is presented in Figure III-93.

The electrical efficiencies for these systems are higher than for simple cycles due to the additional electrical output from the steam turbine. The overall fuel utilization, however, is lower due to the assumed condenser loss. While a loss in fuel utilization is inevitable, various design arrangements can result in somewhat different results. For this study, the gas turbine exhaust is divided into a process heat boiler and a boiler to provide steam for the steam turbine. The steam turbine expands to a condenser and the rest of the steam is used in the process. This approach is convenient for performance and cost calculations and is representative of the other methods.

Estimated Cost

The combined cycle cost was built up using the costs of the components of the system. The current gas turbine cost for its output was used along with the cost of the heat recovery boiler and the cost of the steam turbine-generator-condenser. Installation costs can be determined in the same way, and the overall costs are presented in Figure III-94. The steam turbine generator portion of the system estimated costs are included in Figure III-95 based on the total system output. The corresponding estimated costs for the gas turbine generator and heat recovery boiler are presented in Figure III-96.

The estimated and maintenance cost of 2 mil/KWH was determined by proportioning the output from the gas turbine at its operating cost (2.5 mils/KWH) and the steam turbine generator at its power output (0.6 mils/KWH).

Emissions

Emissions for the combined cycle are the same as for the comparable simple gas turbine in terms of the heating value of the fuel consumed.

Physical Characteristics

The physical characteristics of current combined cycle systems are presented in Table III-38.

TABLE 38 - CURRENT - COMBINED CYCLE
PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Petroleum Distillate	.2	16	90

ADVANCED COMBINED CYCLE-PETROLEUM OR COAL-DERIVED BOILER FUEL

Conversion System Description

The advanced combined cycle (energy conversion system number 23 or 24) consists of a combination of an advanced gas turbine (energy conversion system number 12 or 13) and a current technology steam turbine. The advanced gas turbine directly fired by petroleum or coal-derived boiler fuel is described in this Volume III and the application of the current technology steam turbine is discussed in the preceeding section. The principal consideration in the combination of these two elements is the choice between electrical generation efficiency and the quantity of process heat.

Performance Characteristics

Figure III-97 presents the performance of the advanced combined cycle for three design options varying the proportion of recovered heat.

Estimated Cost

The estimated costs of the advanced combined cycle conversion systems were developed from the estimated costs of the principal elements reported elsewhere in this Volume III. The estimated costs are presented in Figure III-98 for the three design options selected. Note that there is a small variation in electrical output as the size of the steam turbine varies from 5 to 10 megawatts. However, the estimated cost variation is minor and a single cost estimate was used in the computer calculations.

The estimated operating and maintenance costs of 2.4 mils/KWH was determined by proportioning the output of the gas turbine at its operating and maintenance cost of 2.8 mils/KWH and the output of the steam turbine and its cost of 0.6 mils/KWH.

Emissions

The emissions discharged based on the heating valve of the fuel consumed are the same as for the comparable simple gas turbine.

Physical Characteristics

The physical characteristics of the advanced combined cycle conversion system with petroleum or coal-derived boiler fuel are presented in Table III-39.

TABLE III-39. ADVANCED COMBINED CYCLE PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Petroleum or Coal Derived Boiler Fuel	0.17	9.8	71.

Cogeneration Applicability

The advanced combined cycle embodies a number of important characteristics for cogeneration applications. Combined cycles with either extraction or bypass steam turbines provide flexibility to meet a variety of industrial energy needs. The electrical and thermal energy provided can be varied as process needs vary and combined cycle systems can respond promptly to changes in demand. Generally, combined cycle conversion systems are applicable to the larger size applications (10 MW and above) with emphasis on high electrical to thermal energy ratios. With their high electrical efficiency potential, combined cycles could be particularly attractive in grid connected applications, but they can operate independently, also.

With the modular nature and modest space requirements of the elements of combined cycle systems, they are candidates for retrofit applications and additions to meet expanding plant needs can be straightforward.

Combined cycle power plants can operate with any fuels suitable for gas turbine operations.

Future Developments

To achieve widespread cogeneration in the 1985 period, the technical developments described under advanced gas turbines are required.

ADVANCED COMBINED CYCLE-DIRECT COAL FIRED-PRESSURIZED FLUIDIZED BED

Conversion System Description

The advanced combined cycle with direct coal fired combustion (energy conversion system number 25) includes an advanced, direct coal fired gas turbine with a pressurized fluid bed (energy conversion system number 15) and a current technology steam turbine-generator. Both of these elements are described elsewhere in this Volume III.

Performance Characteristics

The performance characteristics of the advanced, direct coal fired, combined cycle are presented in Figure III-99. One design option emphasizing process steam as well as electrical efficiency was selected for this conversion system.

Estimated Costs

The estimated costs of the direct coal fired, advanced, combined cycle are included in Figure III-100. These estimates are based upon the estimated costs of the elements reported elsewhere in this report. The coal-fired pressured fluid bed heat source and gas cleanup system cost estimates are not included in Figure III-100 and are included in Volume IV.

The estimated operating and maintenance cost is 2.6 mils/KWH based on the direct coal fired gas turbine and the steam turbine operating and maintenance estimated costs.

Emissions

The exhaust emissions based on the heating value of the fuel consumed are the same as the emissions from the direct coal fired gas turbine with the pressurized fluidized bed.

Physical Characteristics

The physical characteristics of the direct coal fired combined cycle are presented in Table III-40 except for the pressurized fluidized bed and gas cleanup system. The physical characteristics of these elements are reported in Volume IV.

TABLE III-40. ADVANCED COMBINED CYCLE-DIRECT COAL FIRED PHYSICAL CHARACTERISTICS

Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal - Pressurized Fluidized Bed	0.17	9.8	71.

Cogeneration Applicability

The direct coal fired combined cycle conversion system offers the flexibility of the combined cycle with the applicability of the direct coal fired gas turbine.

Future Developments

The technical developments described under direct coal-fired advanced gas turbines are required to achieve widespread cogeneration in 1985-2000.

ADVANCED COMBINED CYCLE-INDIRECT COAL-FIRED-ATMOSPHERIC FLUIDIZED BED

Conversion System Description

The advanced combined cycle with indirect coal fired atmospheric fluidized bed (energy conversion system number 26) included an advanced indirect coal fired gas turbine with atmospheric fluidized bed (energy conversion system 16) and a current technology steam turbine-generator. Both of these components are described elsewhere in this Volume III.

Performance Characteristics

Figure III-101 presents the performance of the indirect coal-fired advanced gas turbine with atmospheric fluidized bed. These estimates are based on components described elsewhere in this Volume III.

Estimated Cost

The indirect coal-fired advanced combined cycle with atmospheric fluidized bed combustion estimated costs are presented in Figure III-102. These estimates are based upon the estimated costs of the components reported elsewhere in this Volume III. The cost of the atmospheric fluidized bed is not included in Figure III-102, but this information is presented in Volume IV.

The estimated operating and maintenance cost is 1 mil/KWH based on the corresponding costs for the principal components.

Emissions

The emissions, based on the heating value of the fuel consumed, are the same as the emissions from the indirect coal fired gas turbine with an atmospheric fluid bed.

Physical Characteristics

The physical properties of the advanced indirect coal fired combined cycle are presented in Table III-41 except for the atmospheric fluidized bed characteristics which are included in Volume IV.

TABLE III-41. ADVANCED COMBINED CYCLE-INDIRECT COAL FIRED PHYSICAL CHARACTERISTICS

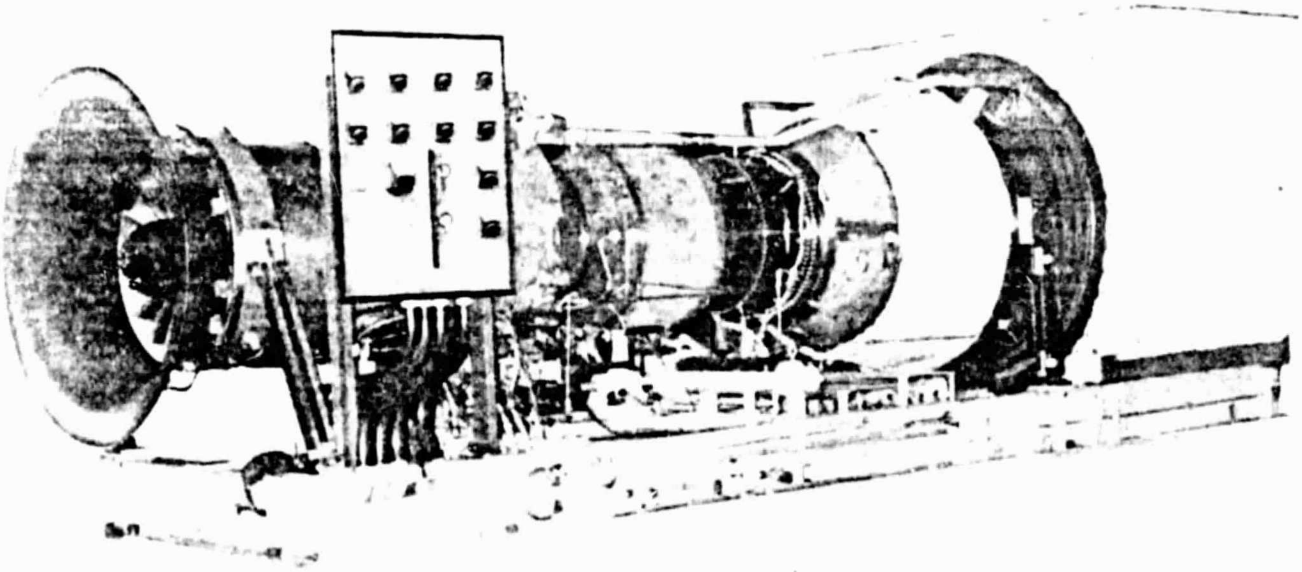
Fuel Type	Footprint Ft^2/KW	Volume Ft^3/KW	Weight #/KW
Coal - Atmospheric Fluidized Bed	0.17	9.8	71.

Cogeneration Applicability

The indirect coal fired advanced combined cycle conversion system offers the flexibility of the combined cycle with the applicability of the gas turbine with the atmospheric fluidized bed coal combustion system.

Future Developments

The technical developments described under indirect coal-fired advanced gas turbines with atmospheric fluidized beds are required to achieve widespread cogeneration in 1985-2000.

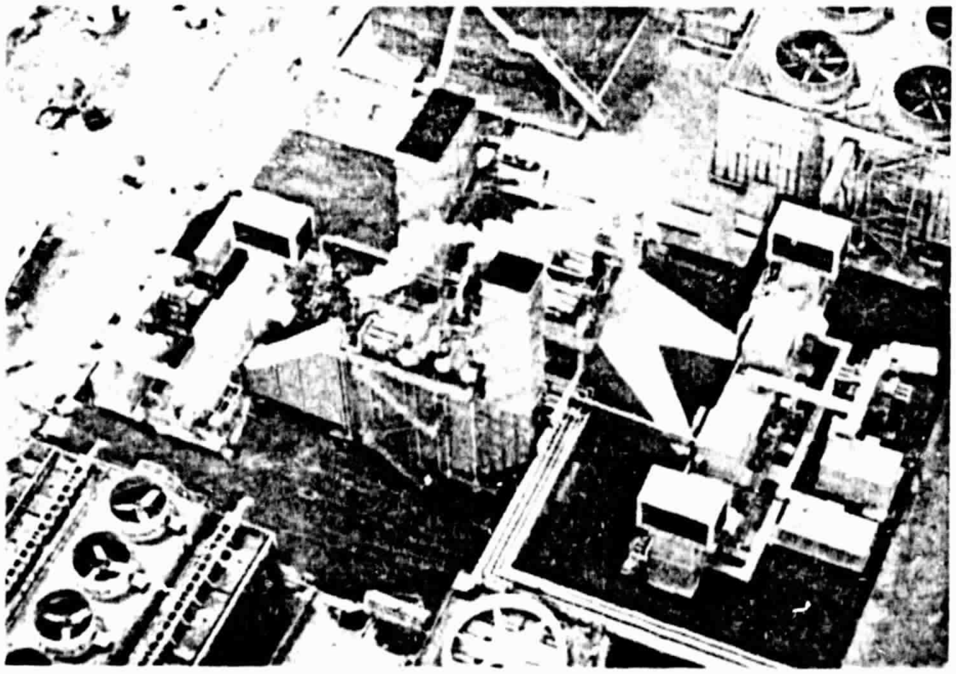


(TC-820)

Figure III-45. 30 MW Industrial Gas Turbine

(CN-19052)

Figure III-46. Gas Turbine Cogeneration Installation at a Chemical Plant



(TC-1031)

Figure III-47. Three Gas Turbine Power Plants with Heat Recovery Steam Generators

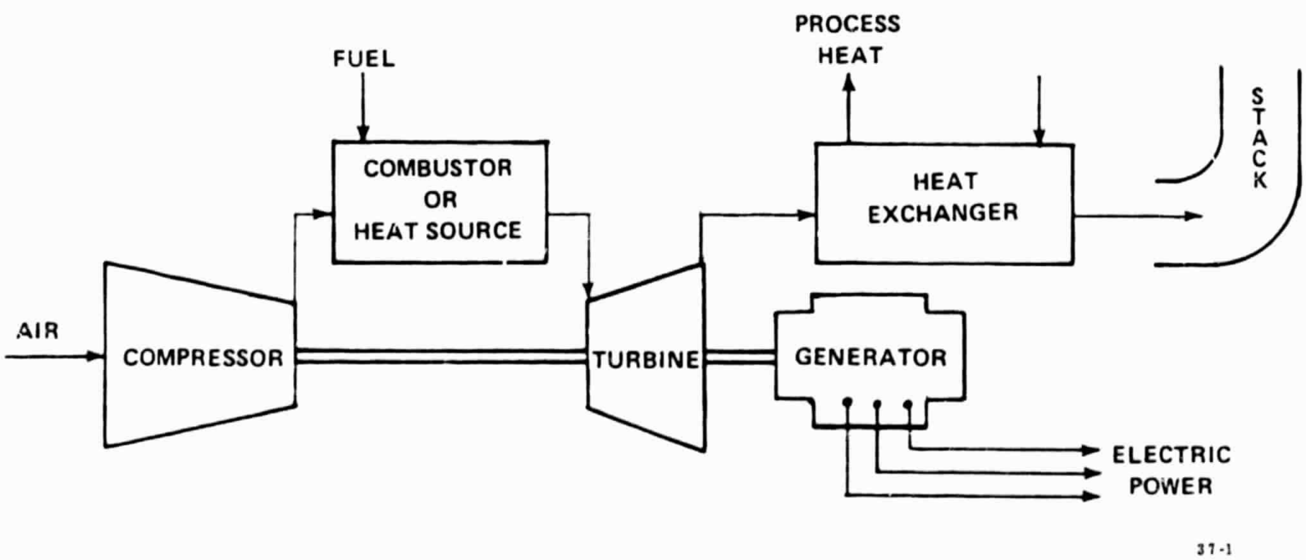


Figure III-48. Gas Turbine Schematic Diagram

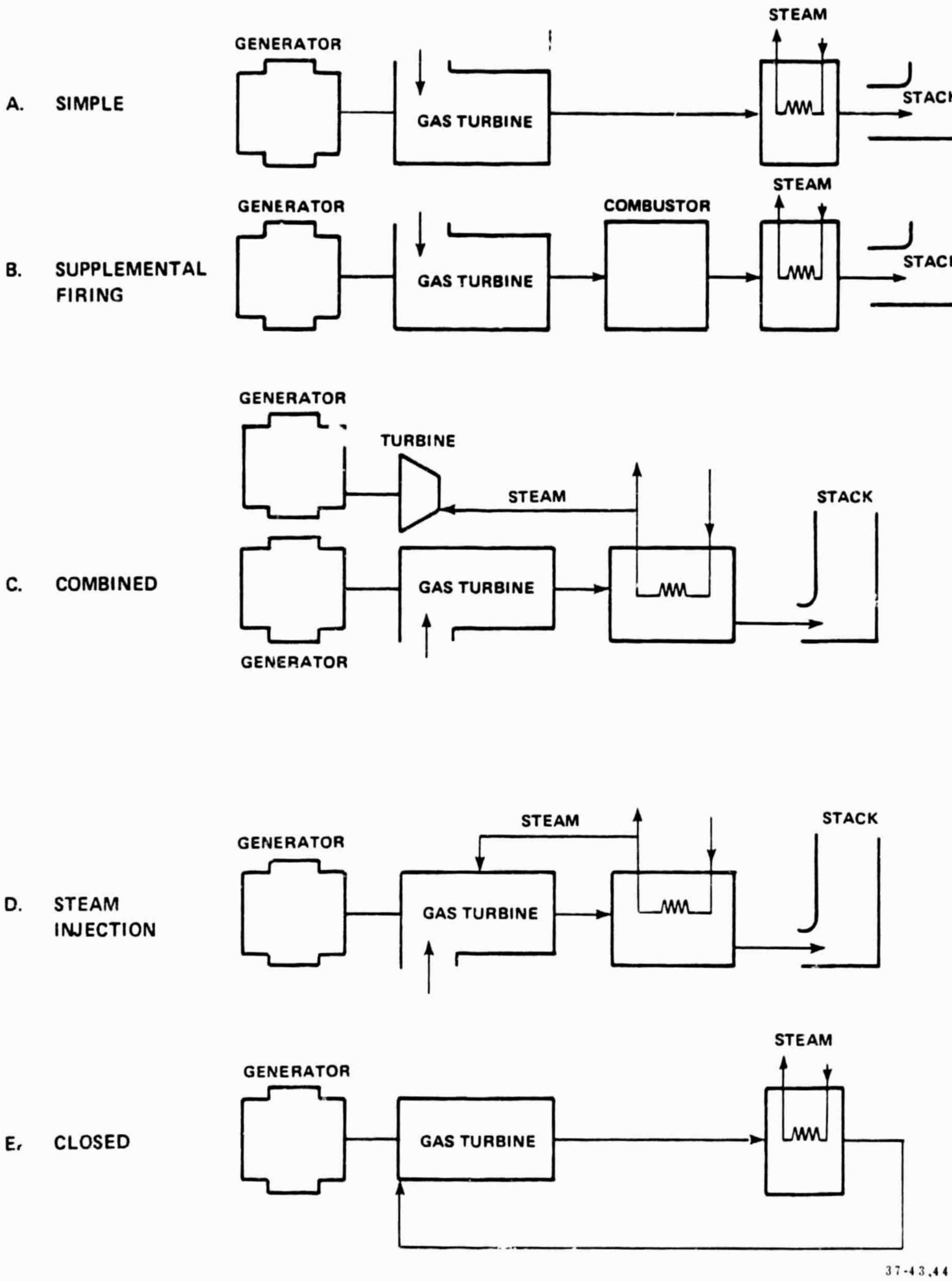


Figure 111-49. Gas Turbine Cogeneration Configurations

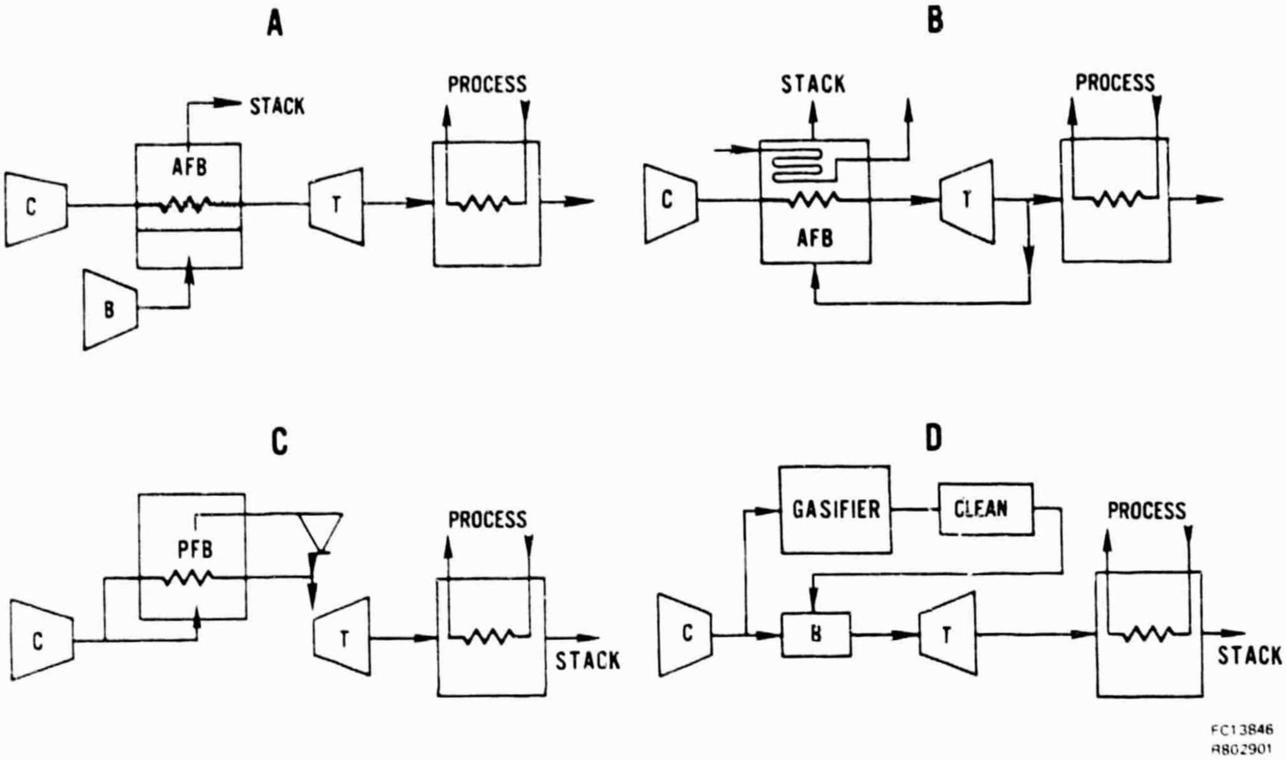


Figure III-50. Coal-Fired Gas Turbine Cogeneration Configurations

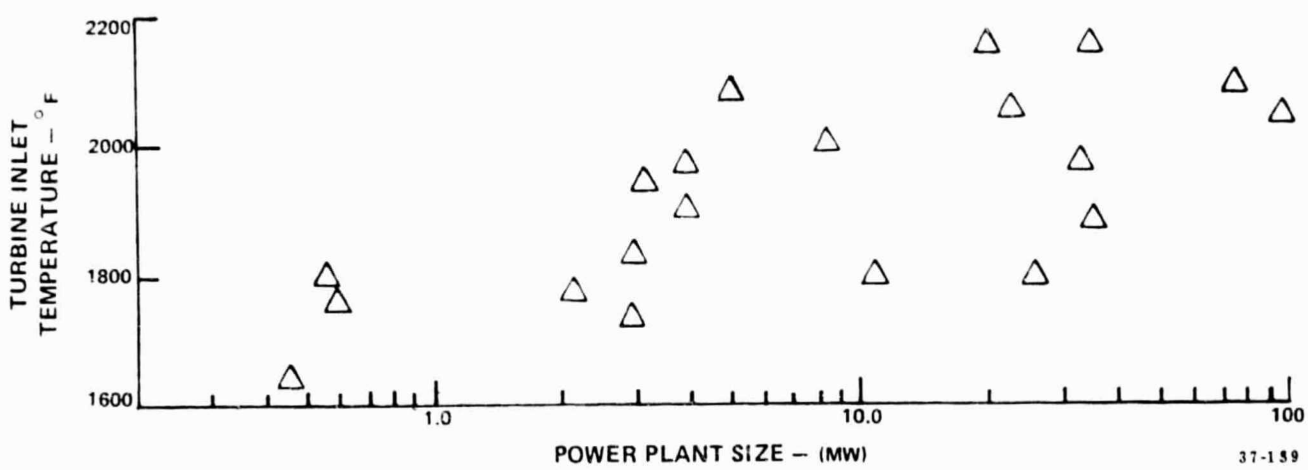


Figure III-51. Current Gas Turbine Maximum Temperature

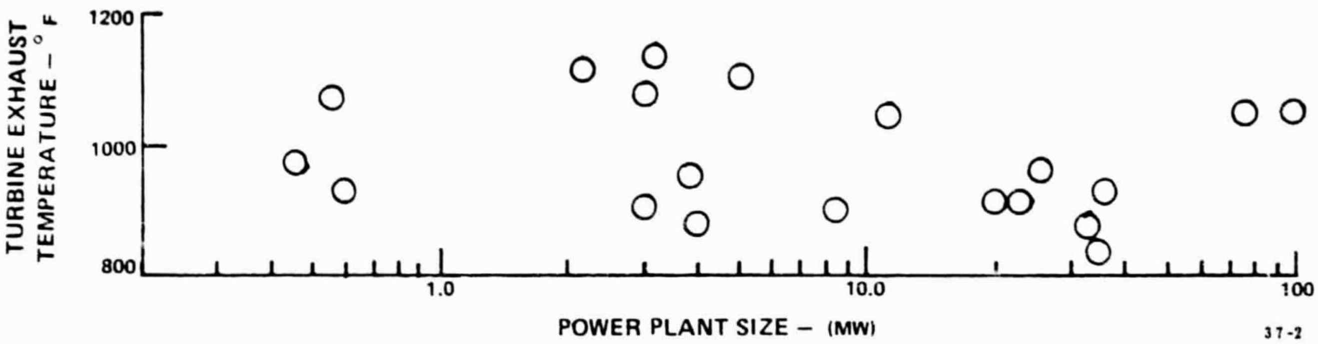


Figure III-52. Current Gas Turbine Discharge Temperature

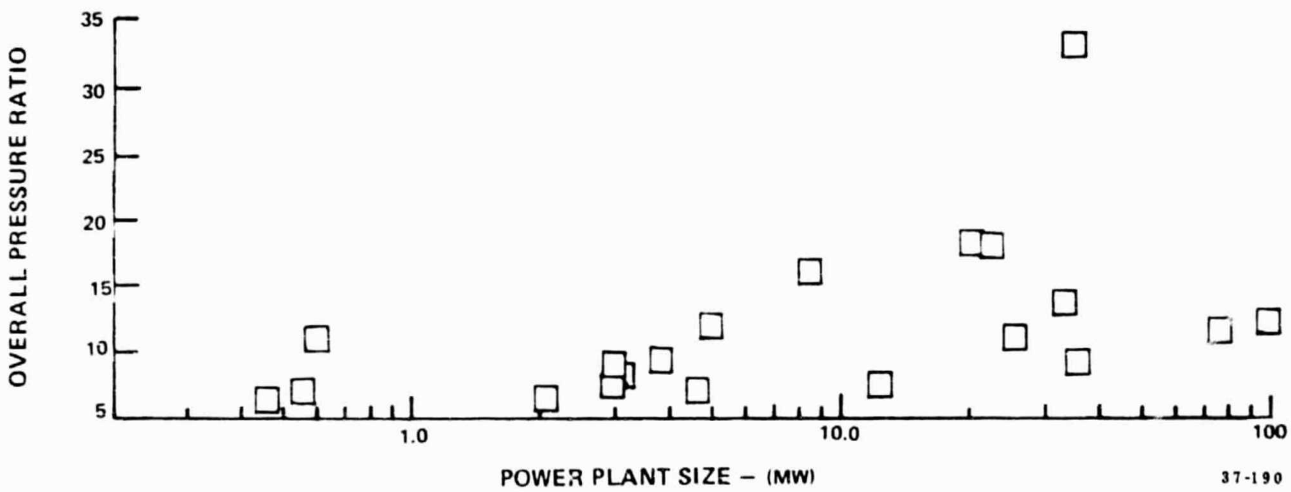


Figure III-53. Current Gas Turbine Pressure Ratio

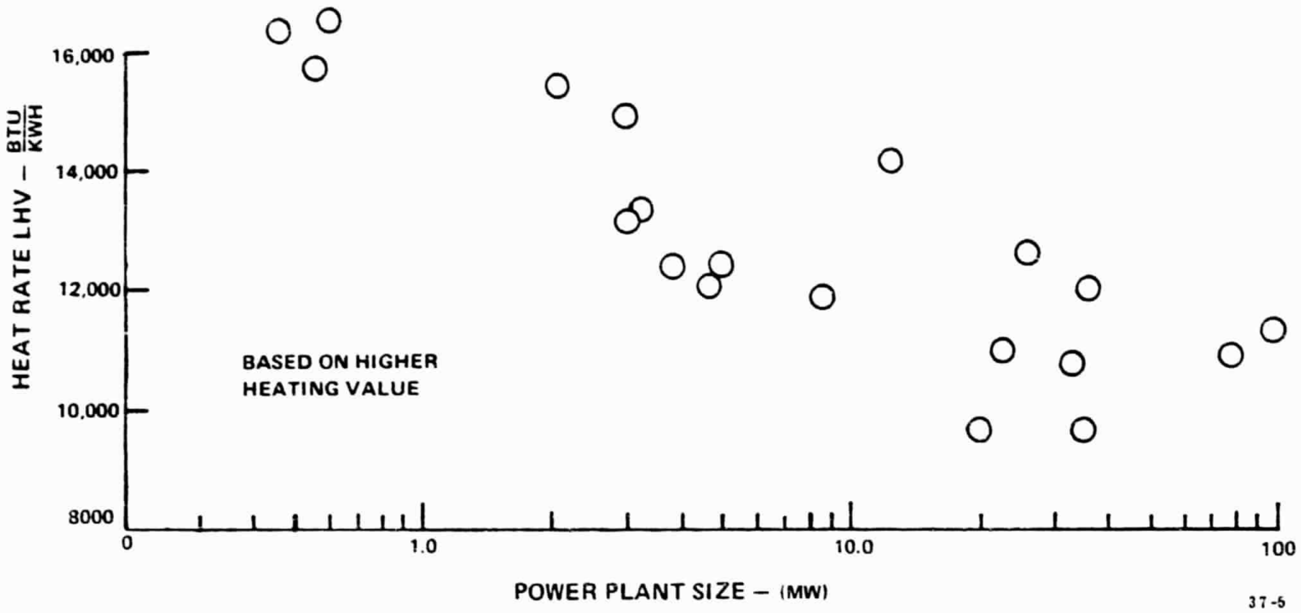


Figure III-54. Current Gas Turbine Efficiency

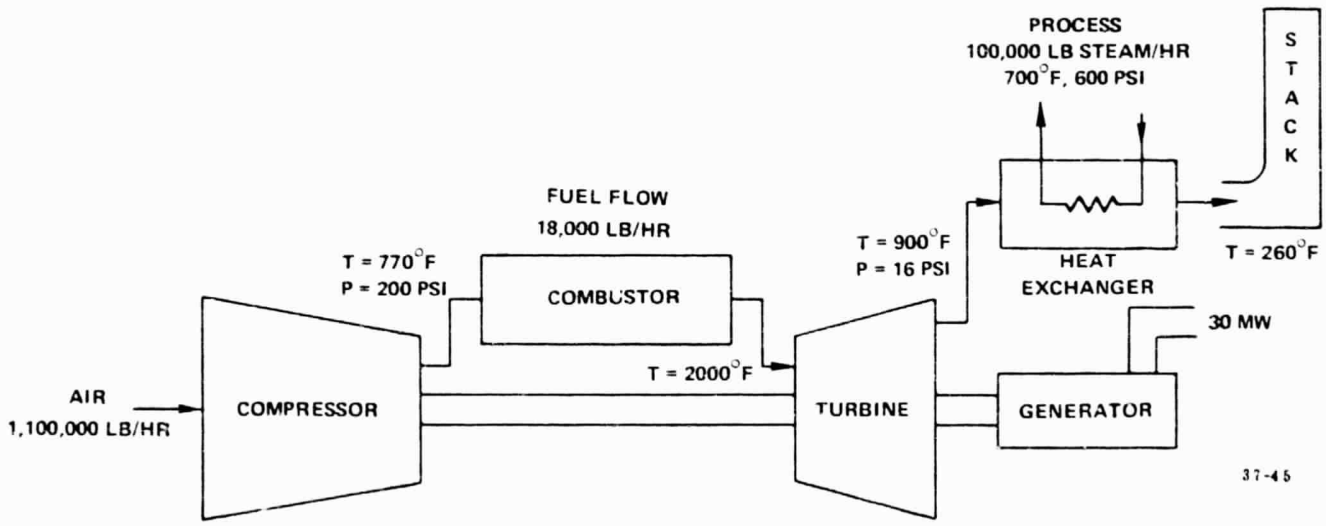


Figure III-55. Typical Current Technology Gas Turbine Cycle

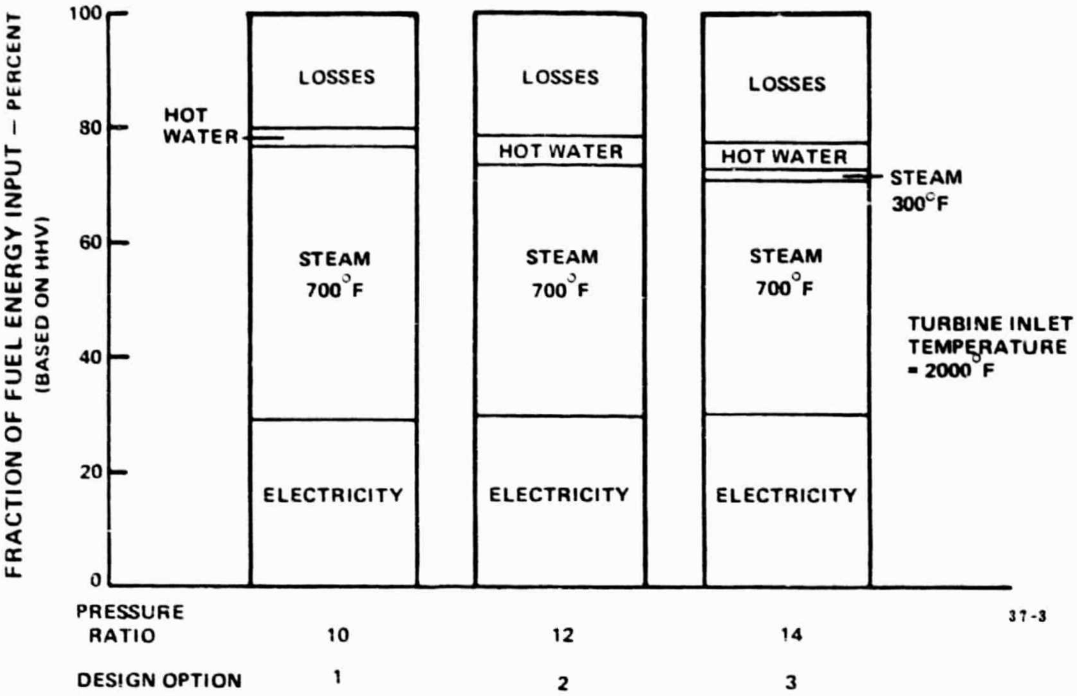


Figure 111-56. Current Gas Turbine Performance with Distillate Fuel

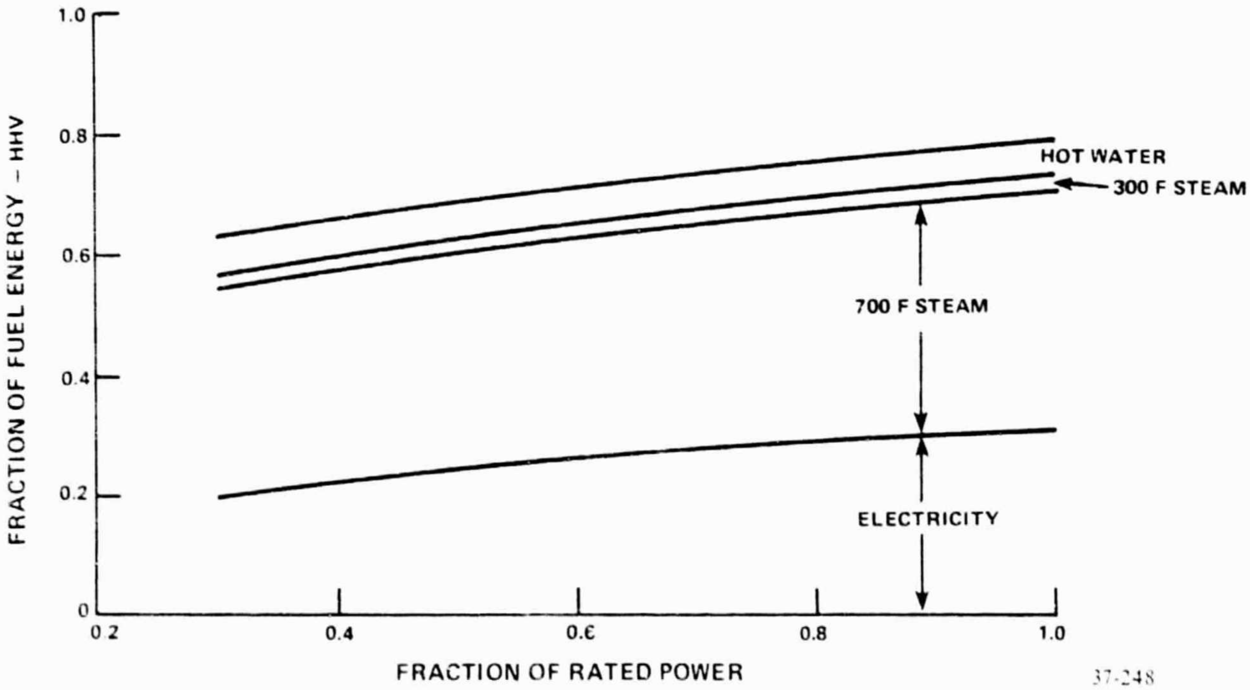


Figure 111-57. Current Gas Turbine Off-Design Performance

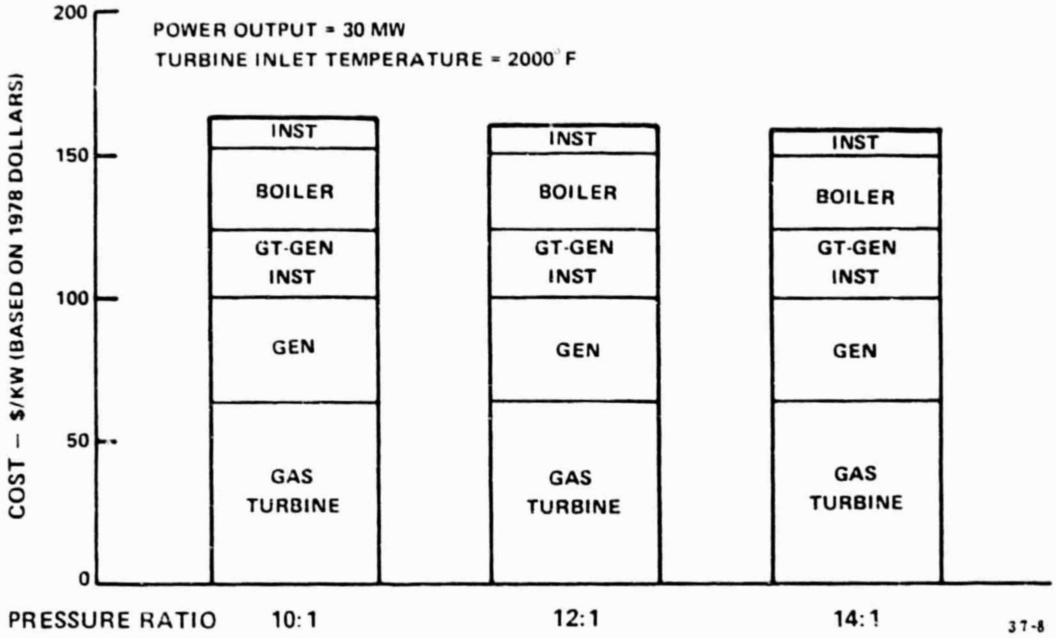


Figure 111-58. Current Gas Turbine Estimated Costs

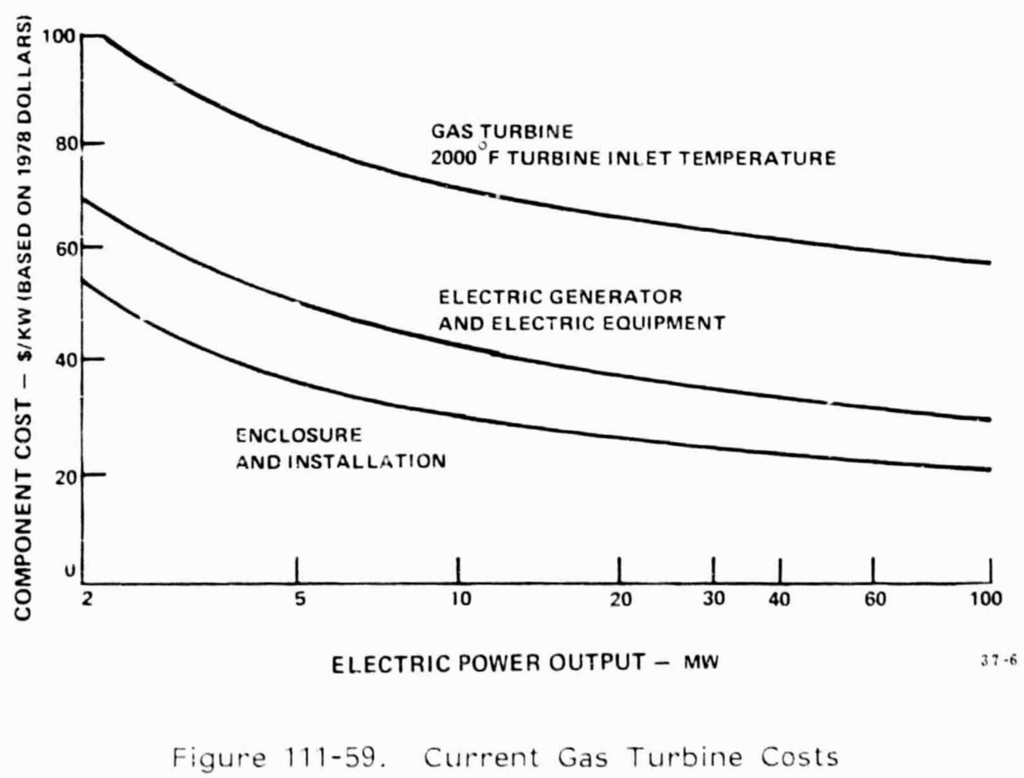


Figure 111-59. Current Gas Turbine Costs

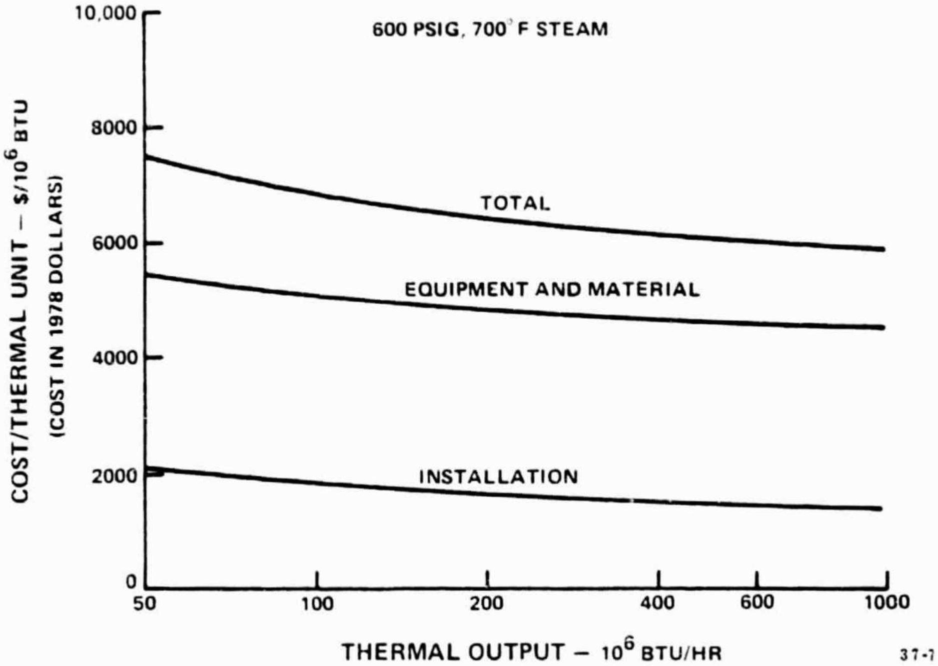


Figure 111-60. Heat Recovery Boiler Cost

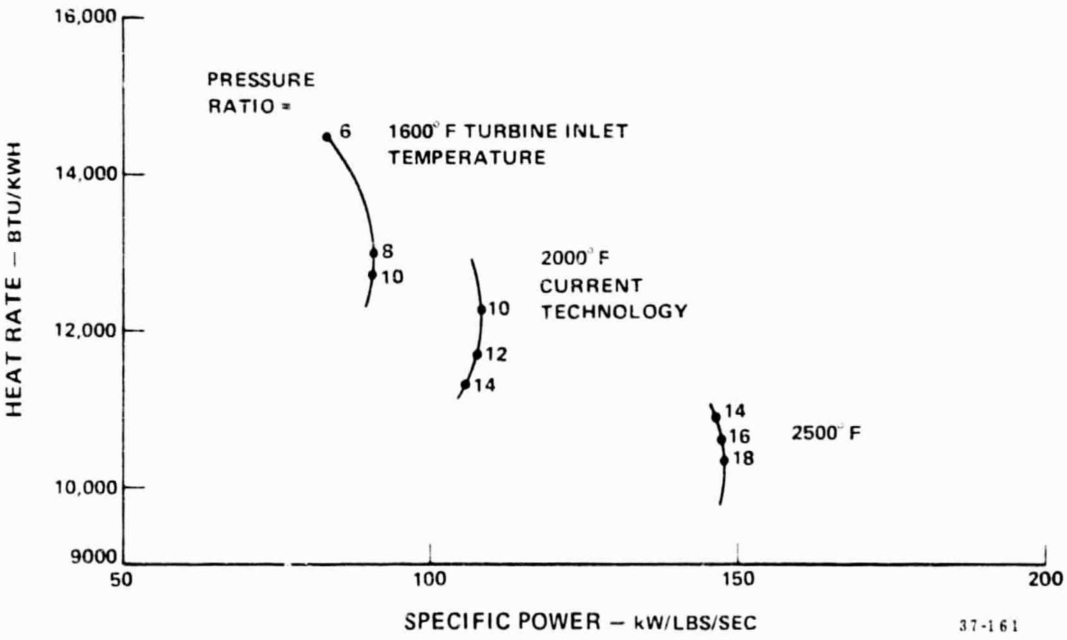


Figure 111-61. Range of Parameters for Current and Advanced Technology Gas Turbines

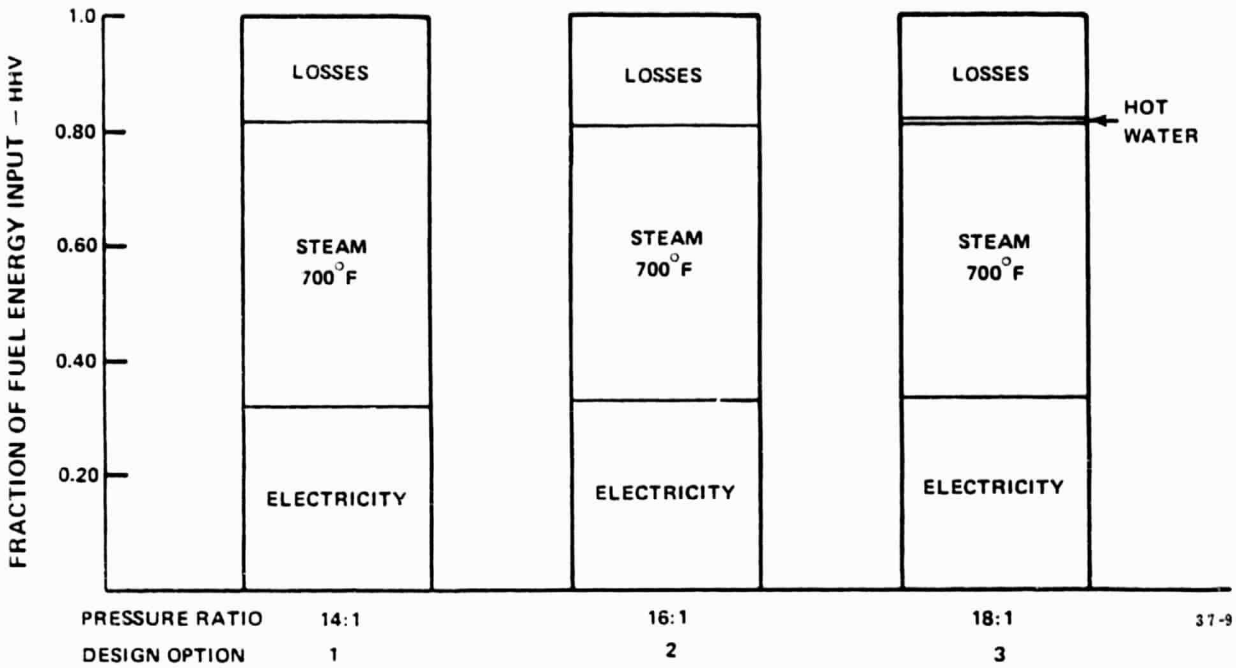


Figure III-62. Advanced Gas Turbine Performance - Petroleum or Coal-Derived Boiler Fuel

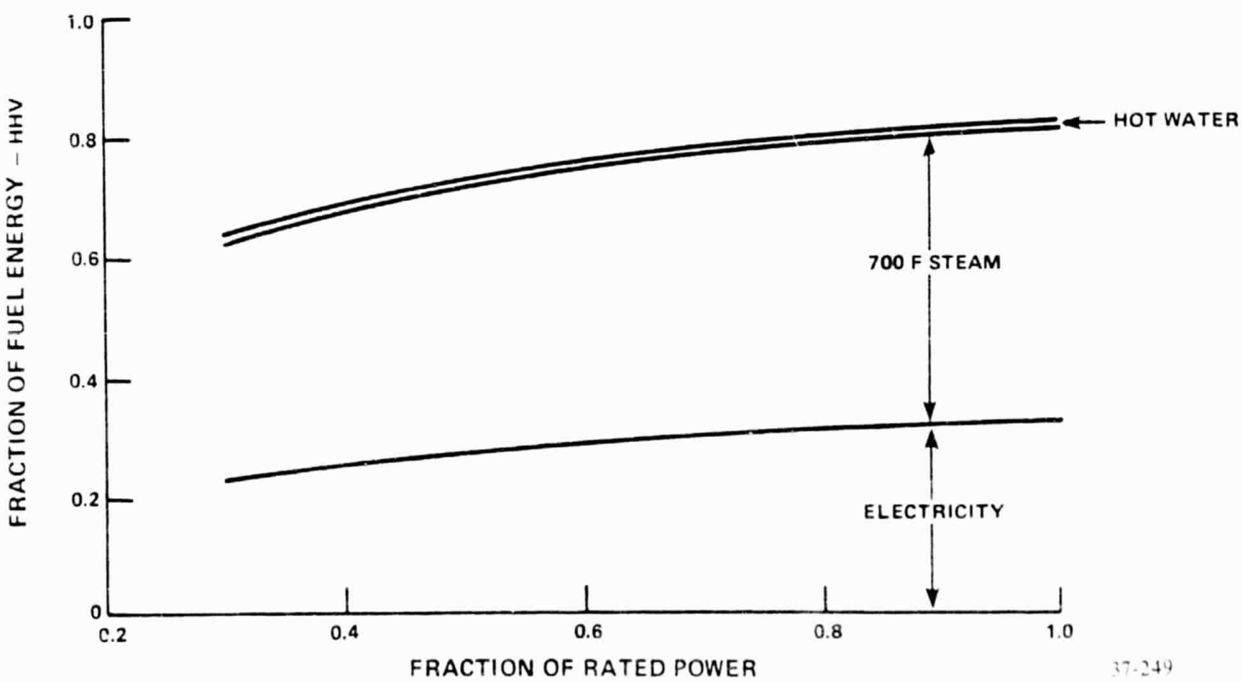


Figure III-63. Advanced Gas Turbine Off-Design Performance - Petroleum or Coal-Derived Boiler Fuel

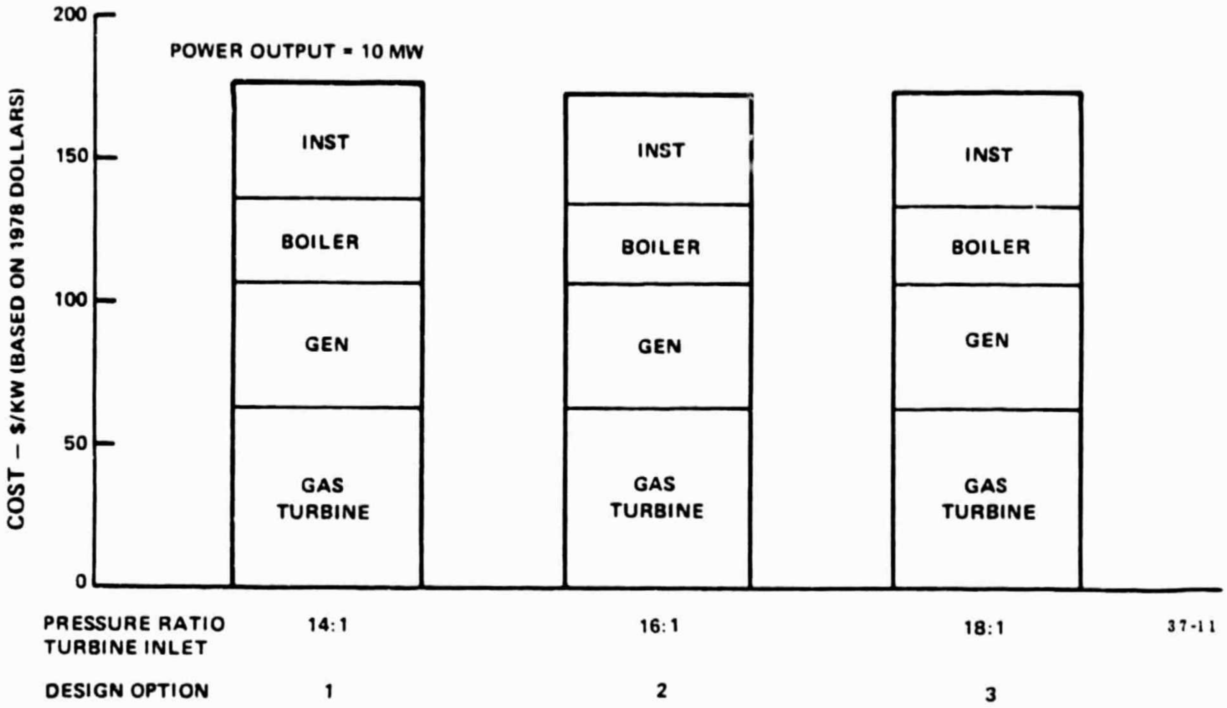


Figure 111-64. Advanced Gas Turbine Estimated Costs - Petroleum or Coal-Derived Boiler Fuel

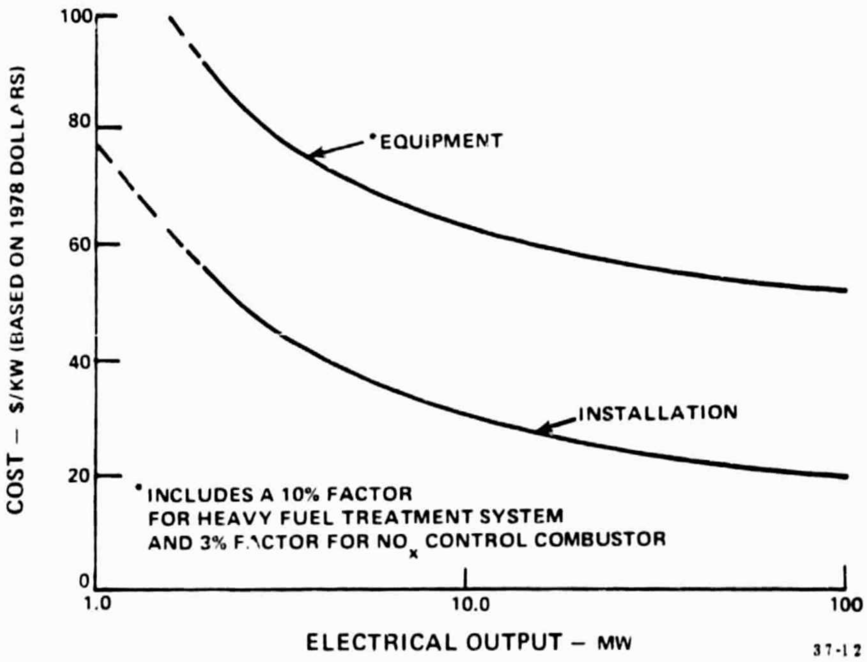


Figure 111-65. Advanced Gas Turbine Engine Estimated Costs - Petroleum or Coal-Derived Boiler Fuel

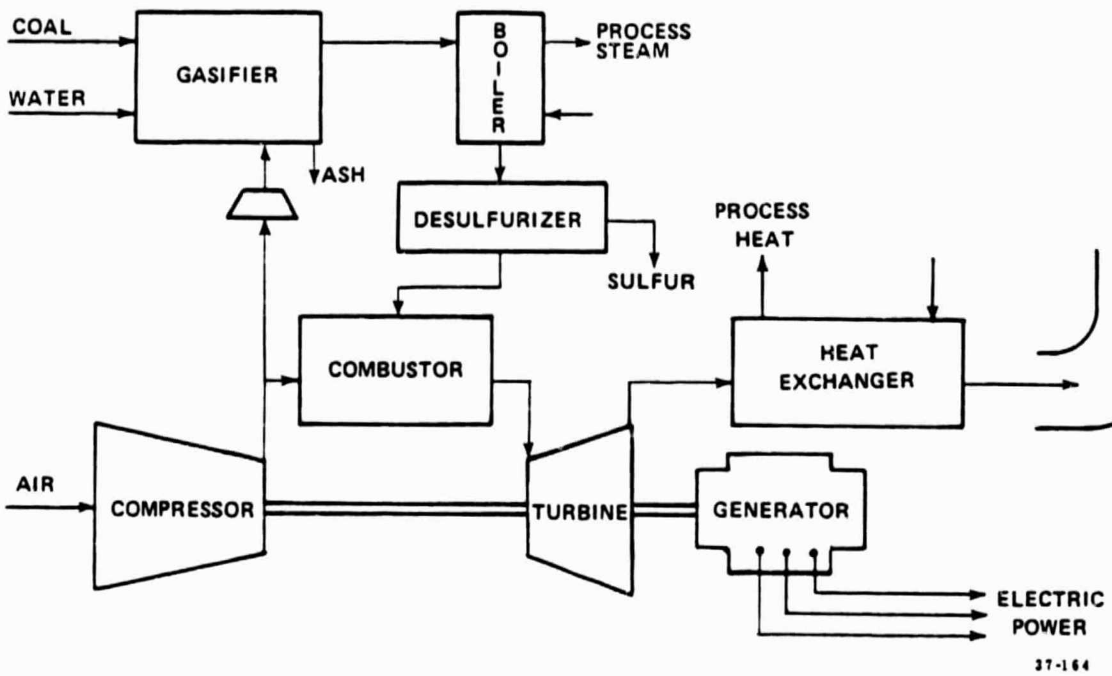


Figure III-66. Schematic Diagram of Advanced Gas Turbine with Coal Gasifier

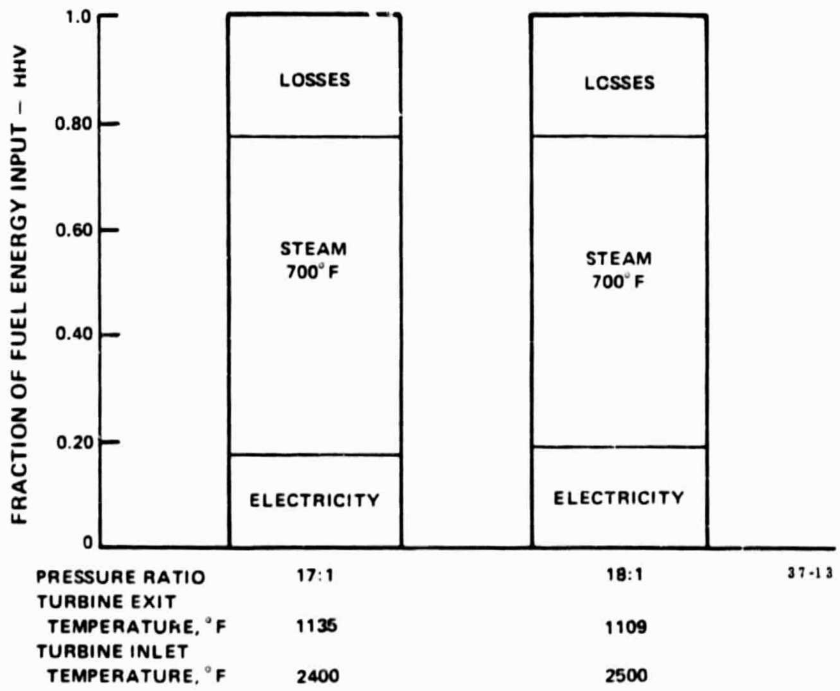


Figure III-67. Advanced Gas Turbine Performance with Gasified Coal

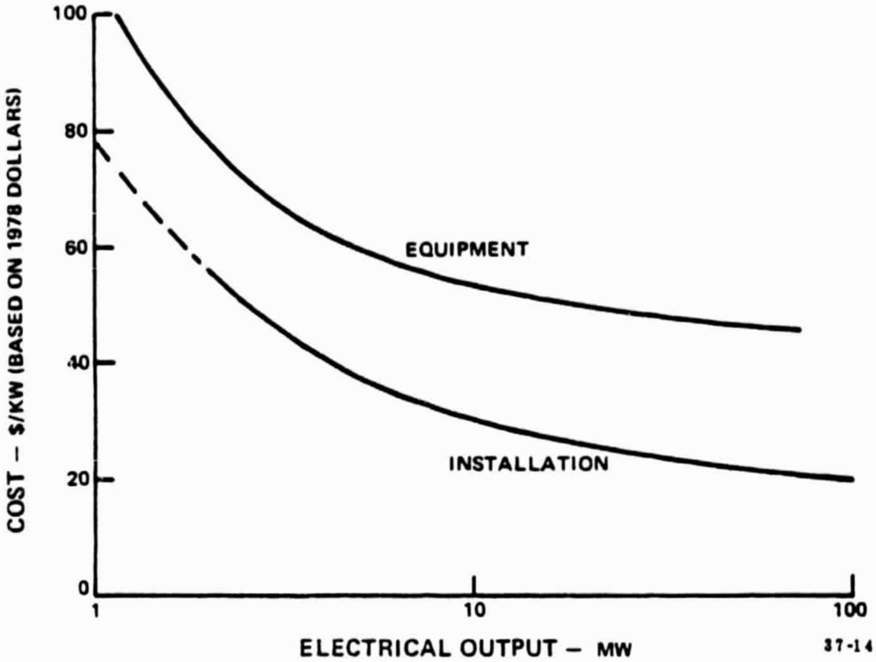


Figure 111-68. Advanced Gas Turbine Engine Estimated Costs - Gasified Coal Fuel

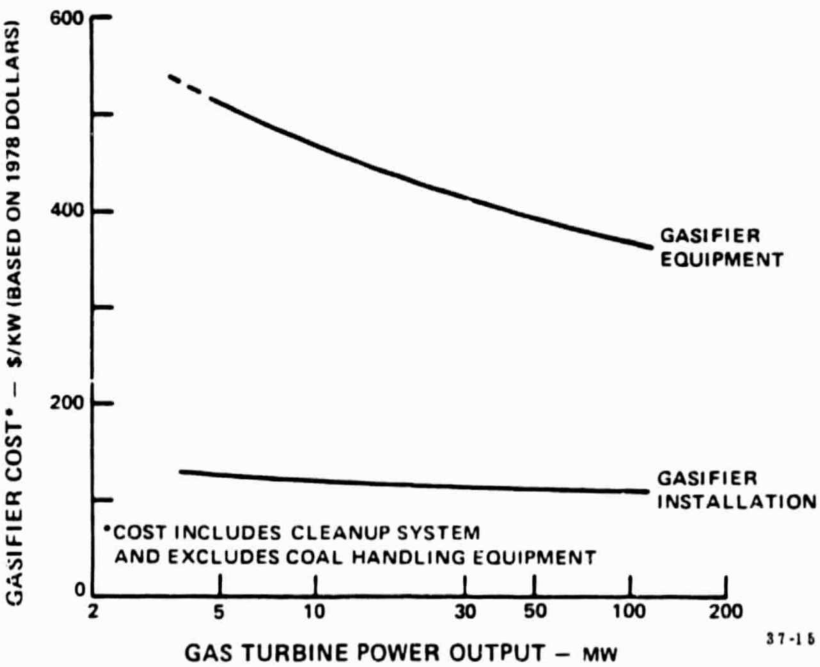


Figure 111-69. Estimated Coal Gasifier Costs

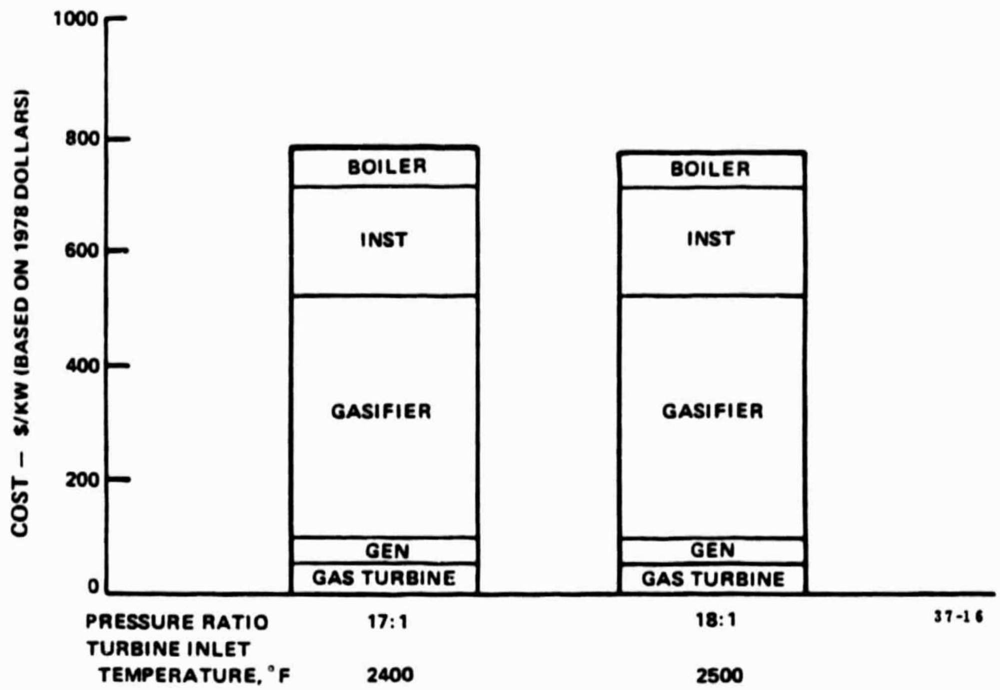


Figure 111-70. Advanced Gas Turbine Estimated Costs - Gasified Coal Fuel

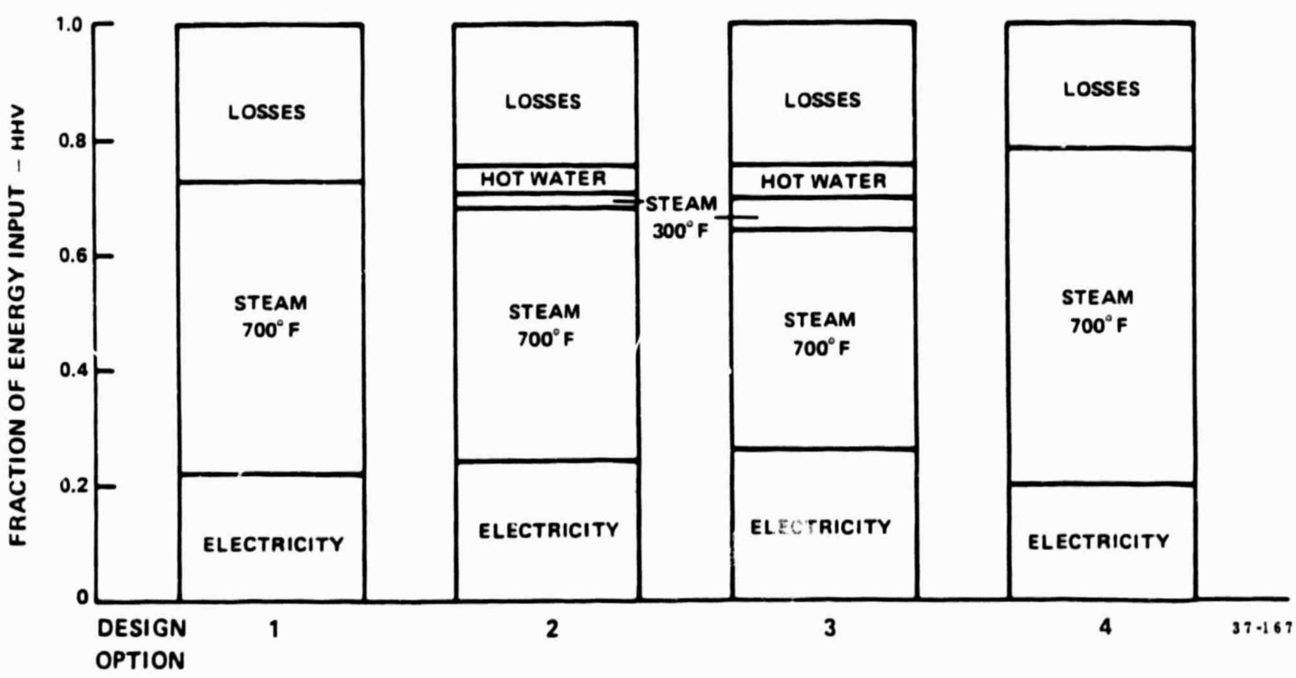


Figure 111-71. Direct Coal-Fired Advanced Gas Turbine Performance

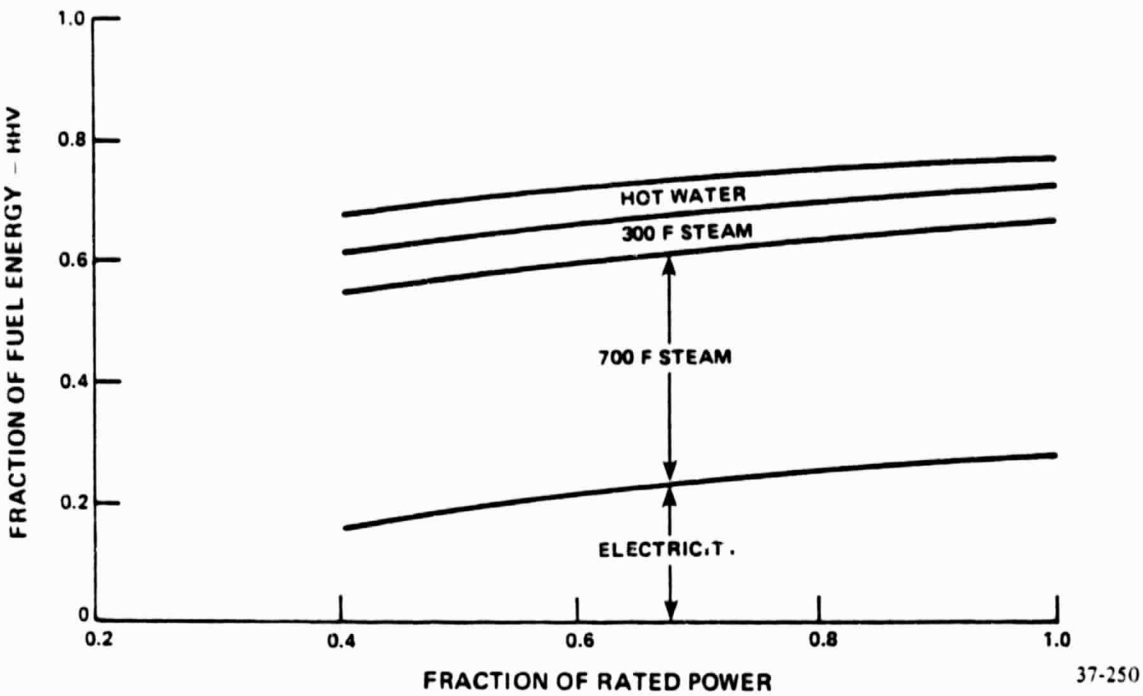


Figure 111-72. Coal-Fired Advanced Gas Turbine Off-Design Performance

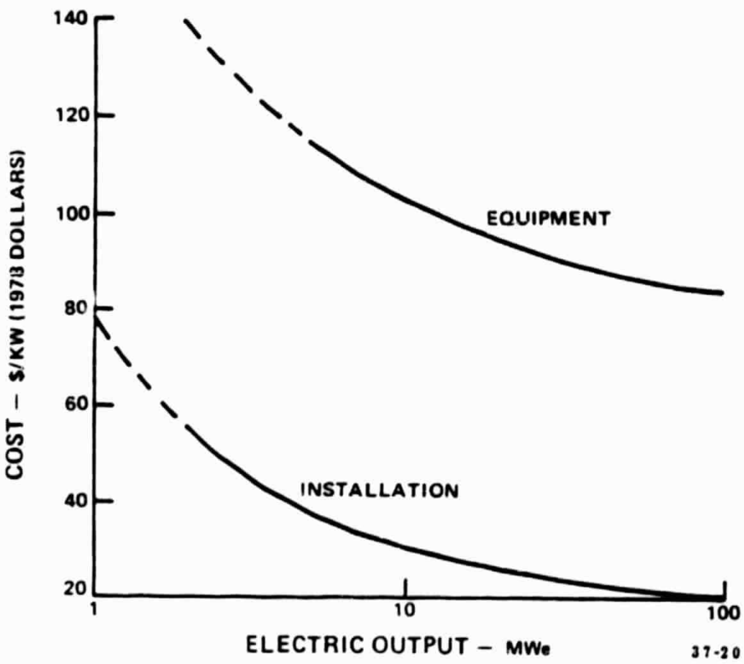


Figure 111-73. Advanced Gas Turbine Engine Estimated Costs - Turbine Inlet Temperature 1500-1600° F

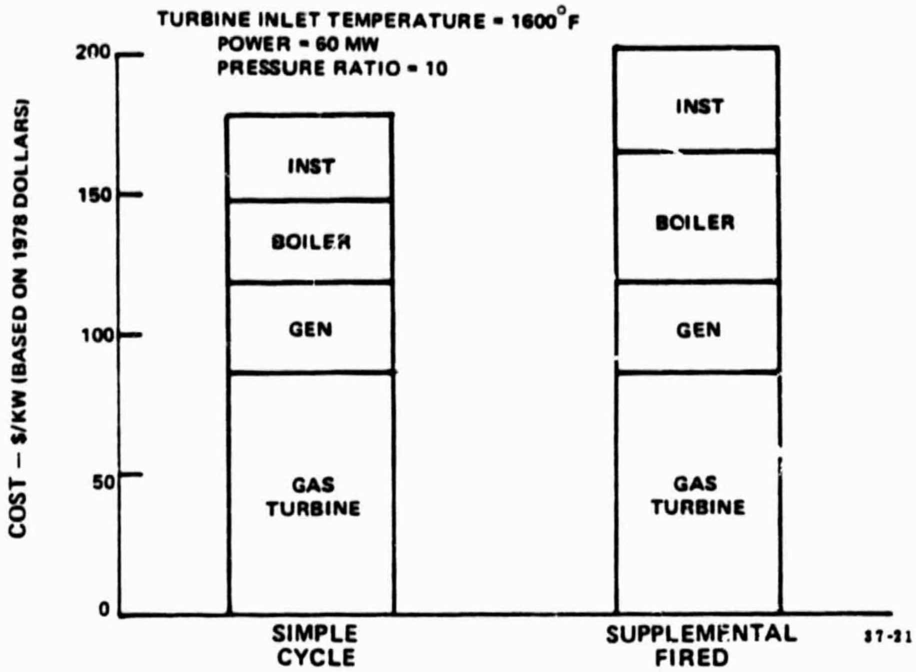


Figure 111-74. Estimated Costs of Direct Coal-Fired Advanced Gas Turbines

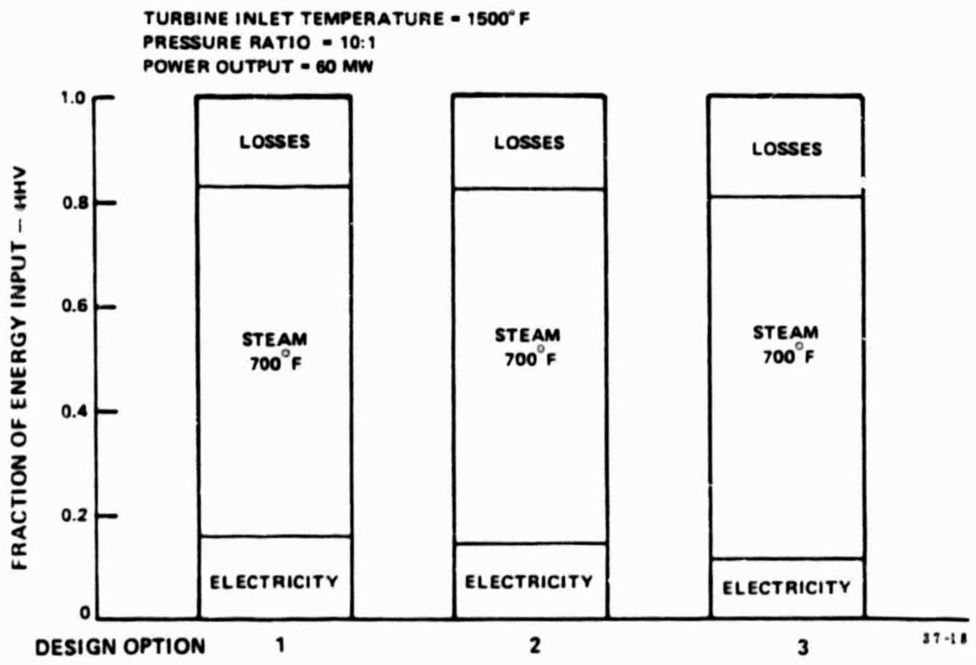


Figure 111-75. Advanced Gas Turbine Performance with Atmospheric Fluid Bed Coal-Fired Heat Source

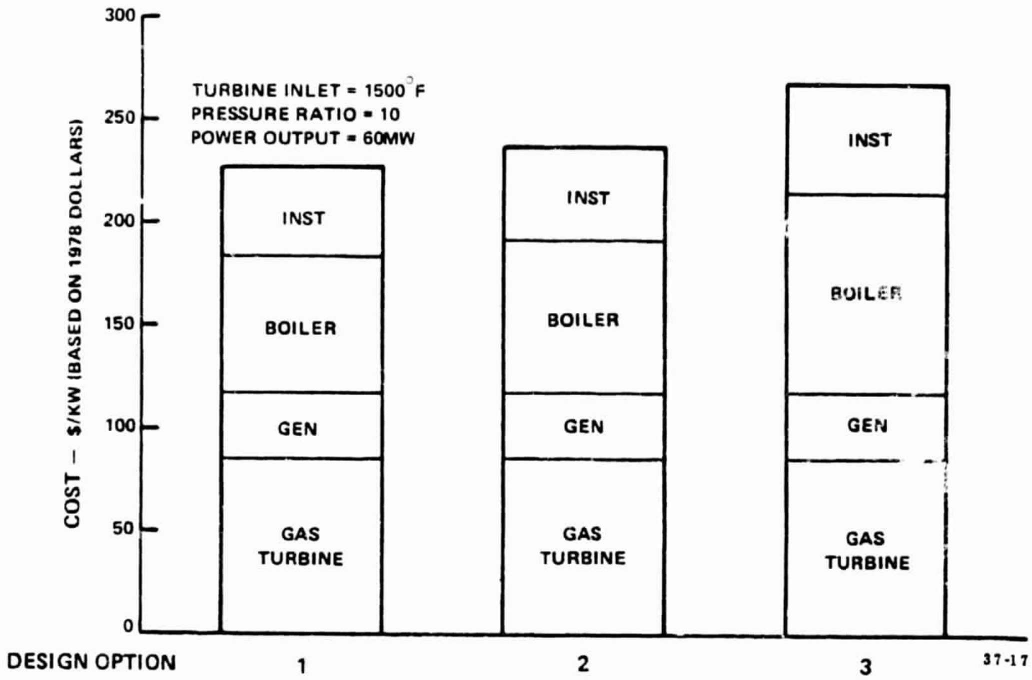


Figure 111-76. Estimated Costs of Advanced Gas Turbine with Atomospheric Fluid Bed Coal-Fired Heat Source

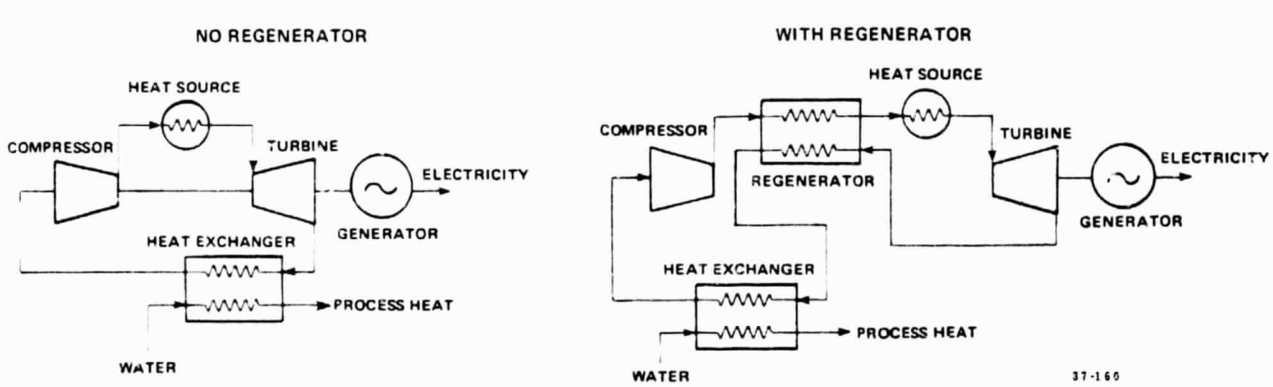


Figure 111-77. Schematic Diagrams of Closed Gas Turbine Cycles With and Without Regeneration

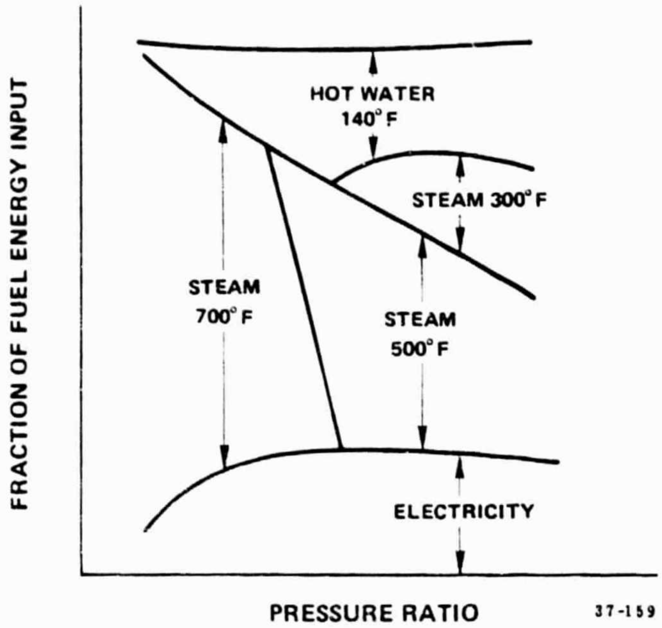


Figure III-78. Pressure Ratio Effect on the Performance of Closed Gas Turbine Without Regenerator

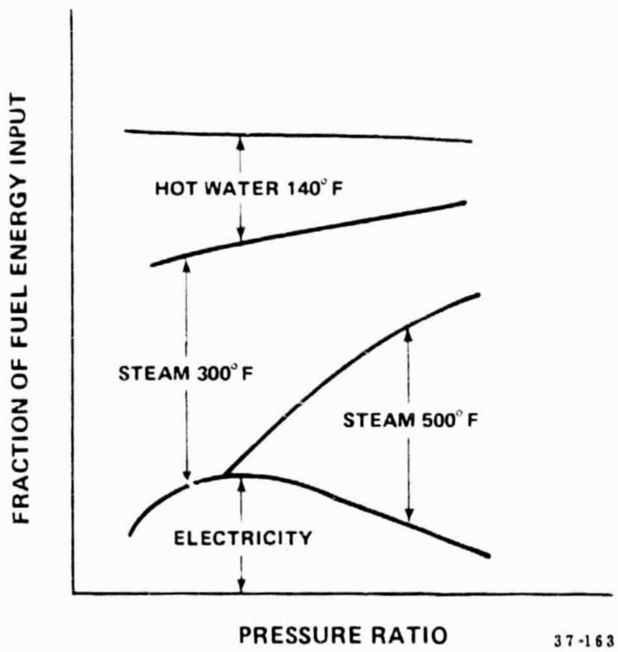


Figure III-79. Pressure Ratio Effect on Performance of Closed Gas Turbine With Regenerator

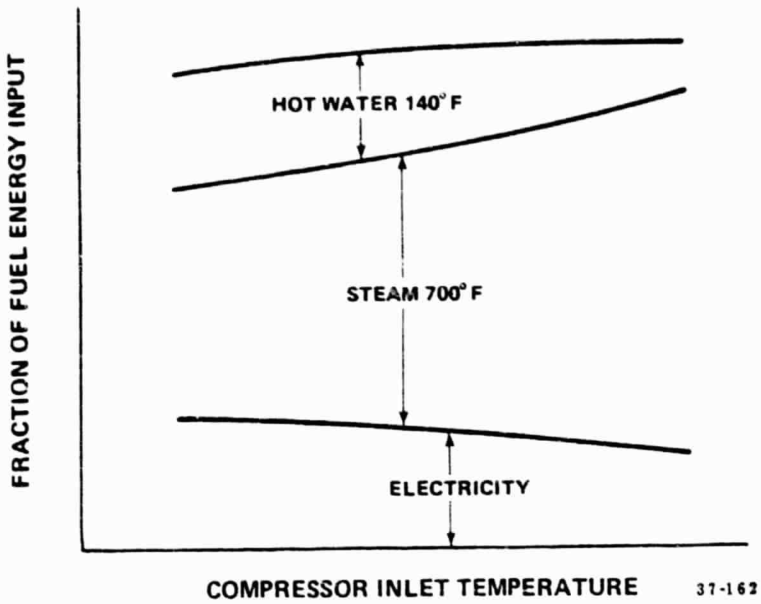


Figure III-80. Compressor Inlet Temperature Effect on Performance for Closed Gas Turbine Without Regenerator

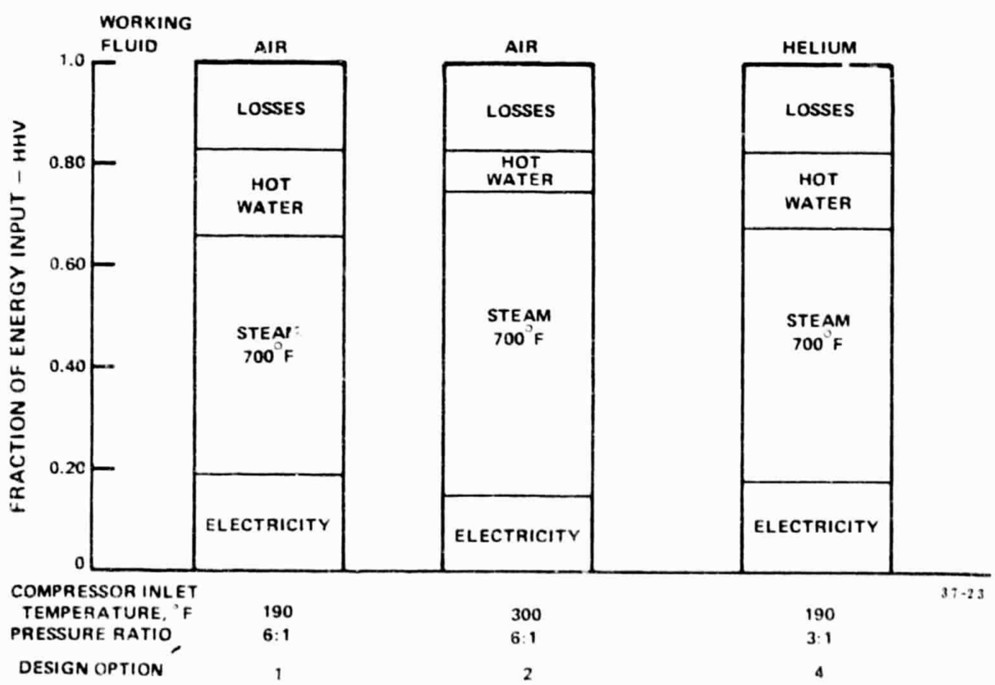


Figure III-81. Closed Cycle Gas Turbine Performance Without Regenerator With Coal-Fired Atmospheric Fluidized Bed Heat Source

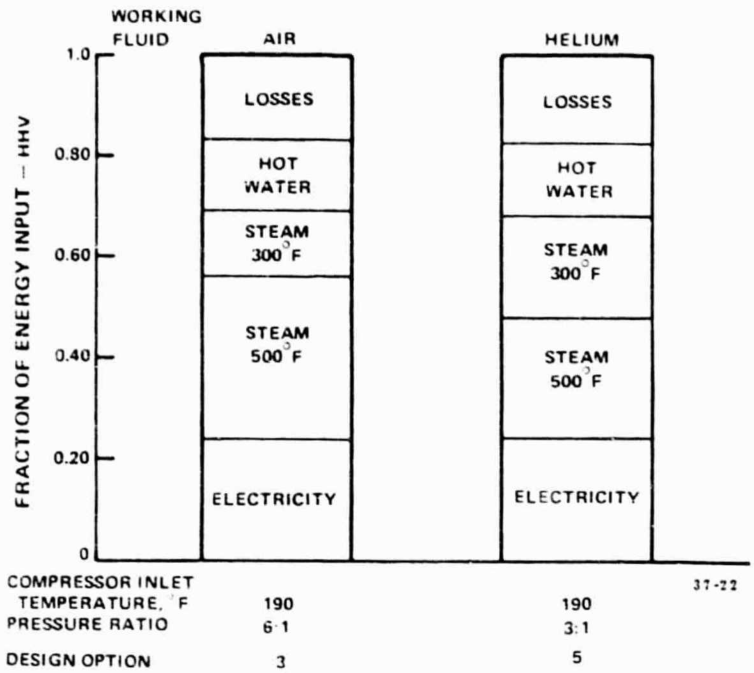


Figure III-82. Closed Cycle Gas Turbine With Regenerator With Coal-Fired Atmospheric Fluidized Bed

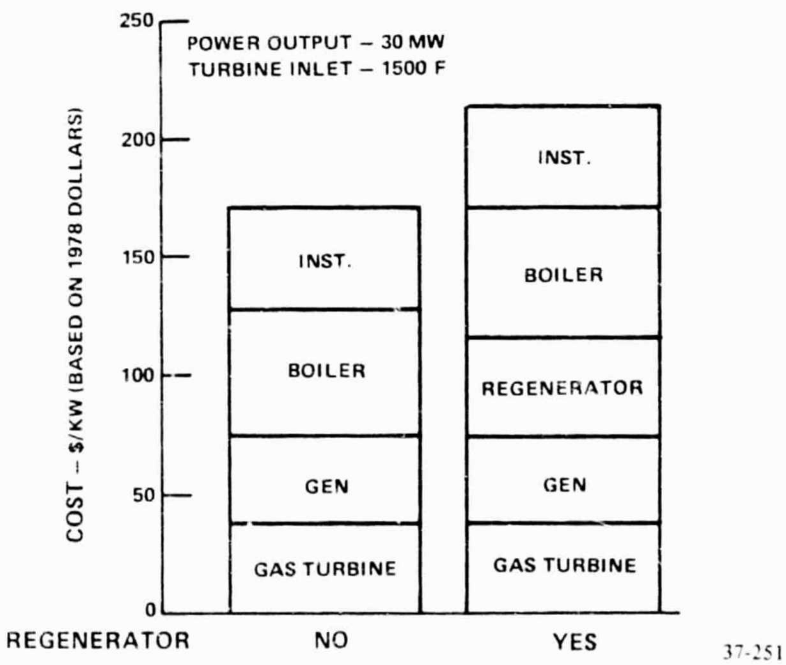


Figure III-83. Estimated Costs of Closed Cycle Gas Turbine With Atmospheric Fluidized Bed Coal-Fired Heat Source

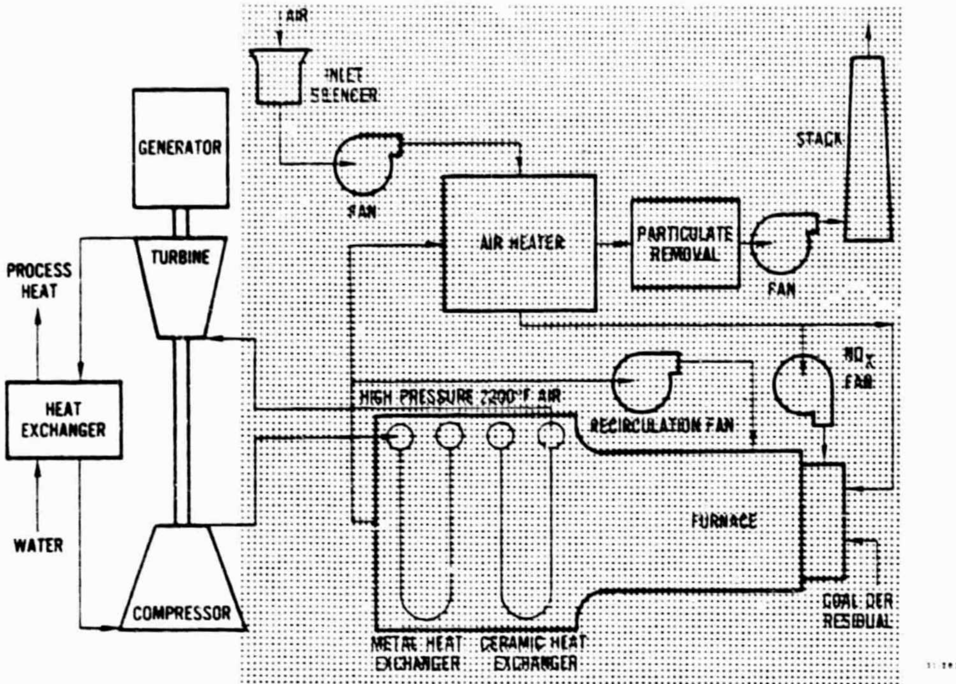


Figure 111-84. Schematic Diagram of Closed Cycle Gas Turbine With High Temperature Oil-Fired Heat Source

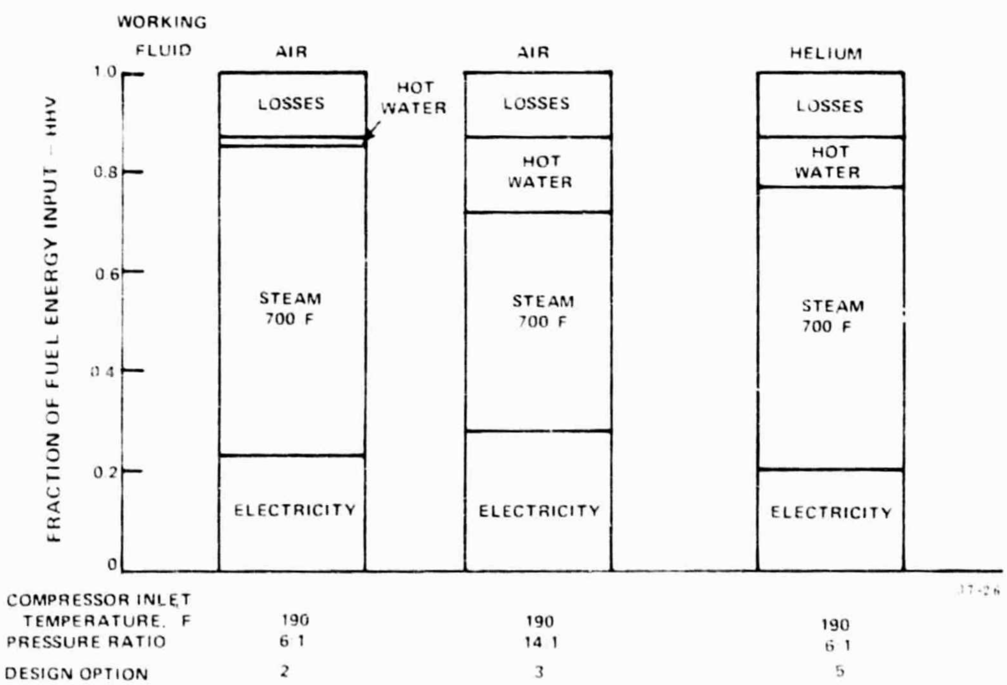


Figure 111-85. Closed Cycle Gas Turbine Performance Without Regenerator With Liquid Fueled Heat Source

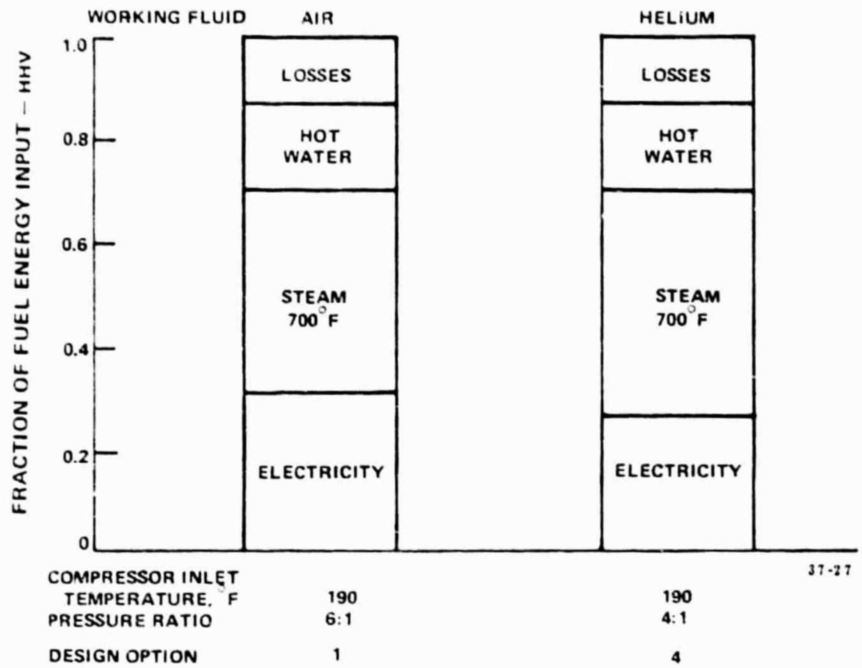


Figure III-86. Closed Gas Turbine Performance With Regenerator With Liquid Fueled Heat Source

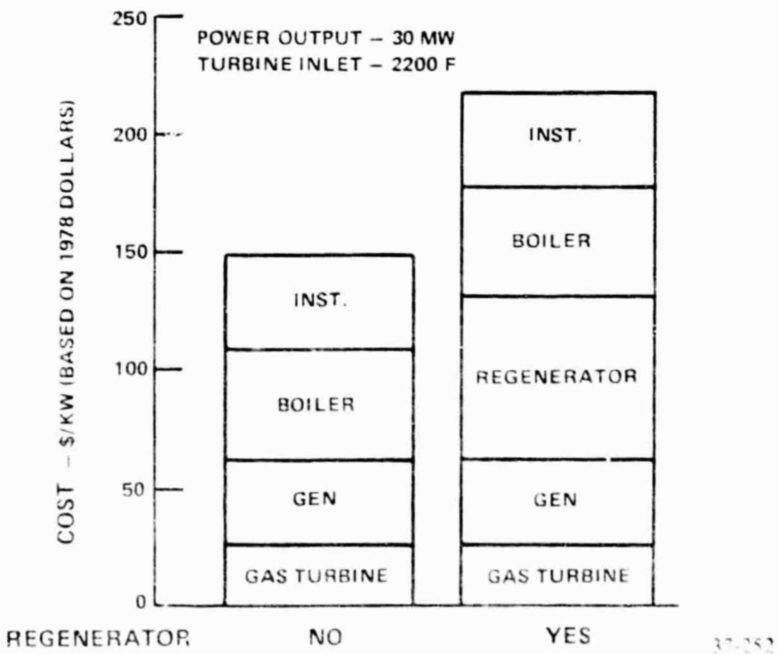


Figure III-87. Estimated Costs of Closed Cycle Gas Turbine With Liquid Fueled High Temperature Heat Source

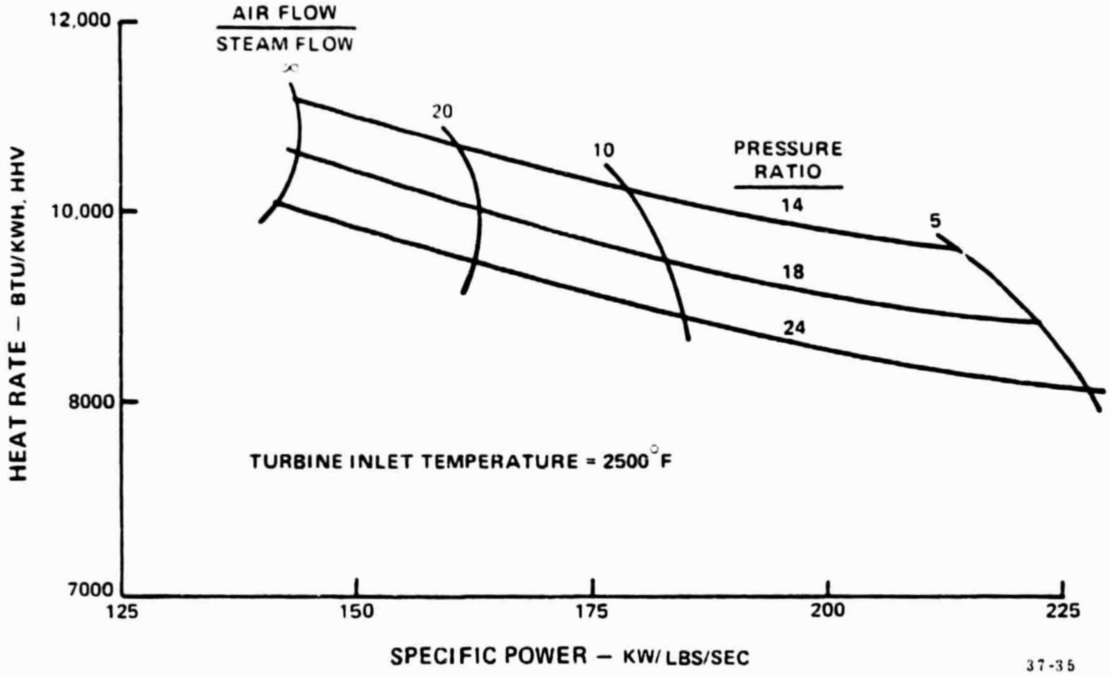


Figure 111-88. Range of Performance Parameters For Advanced Gas Turbines With Steam Injection

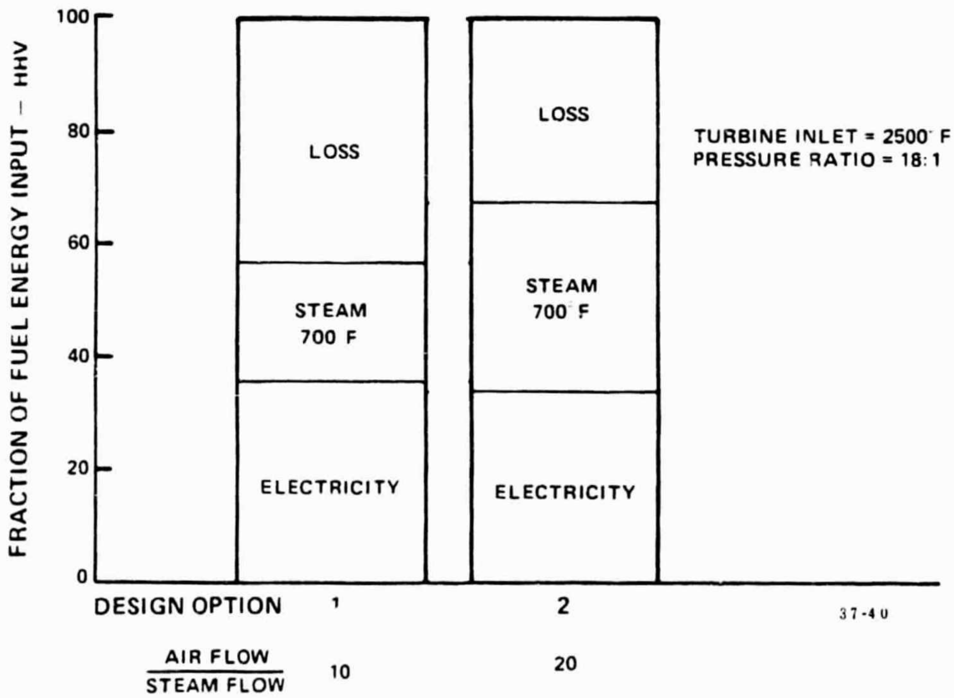


Figure 111-89. Steam Injected Advanced Gas Turbine Performance - Petroleum or Coal-Derived Boiler Fuel

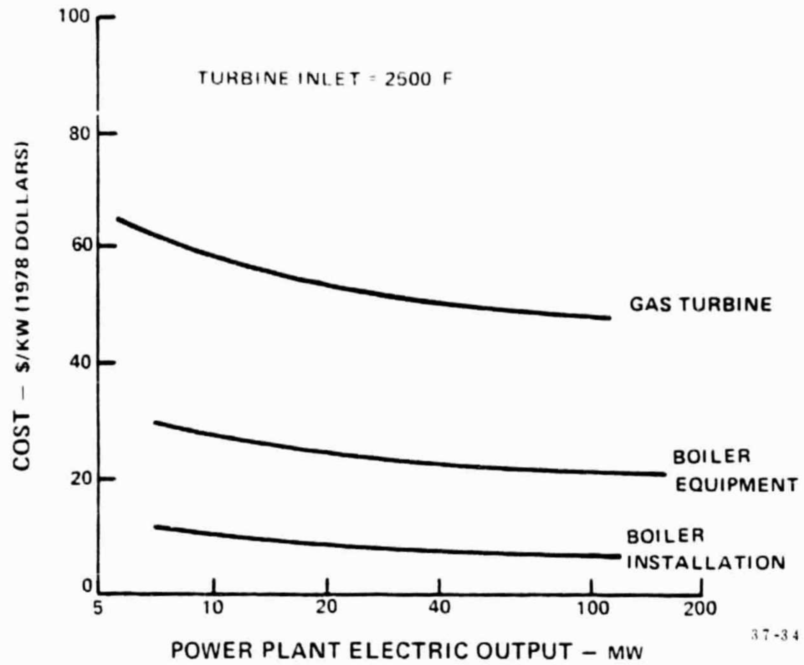


Figure 111-90. Steam Injected Advanced Gas Turbine Estimated Costs - Petroleum or Coal-Derived Boiler Fuel

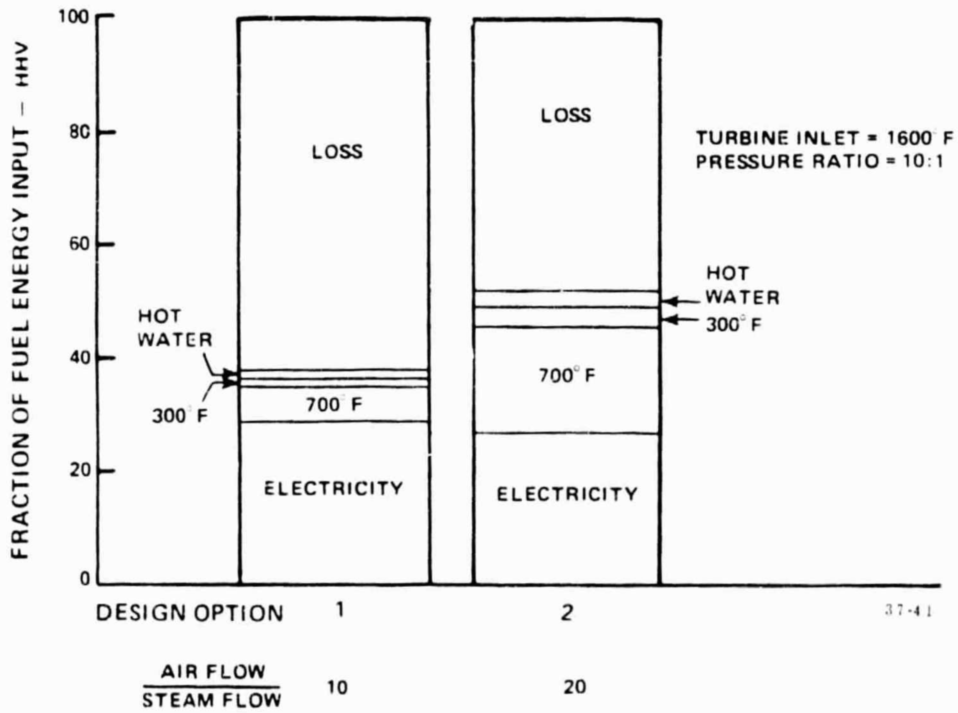


Figure 111-91. Steam Injected Advanced Gas Turbine Performance - Direct Coal-Fired Pressurized Fluidized Bed

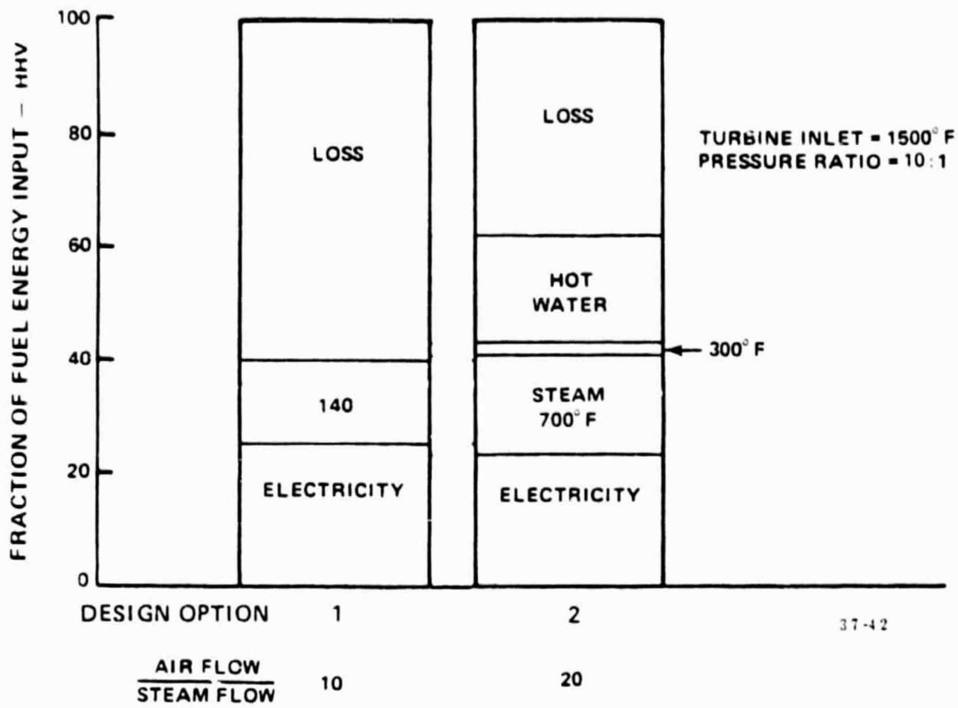


Figure 111-92. Steam Injected Advanced Gas Turbine Performance With Coal-Fired Atmospheric Fluidized Bed Heat Source

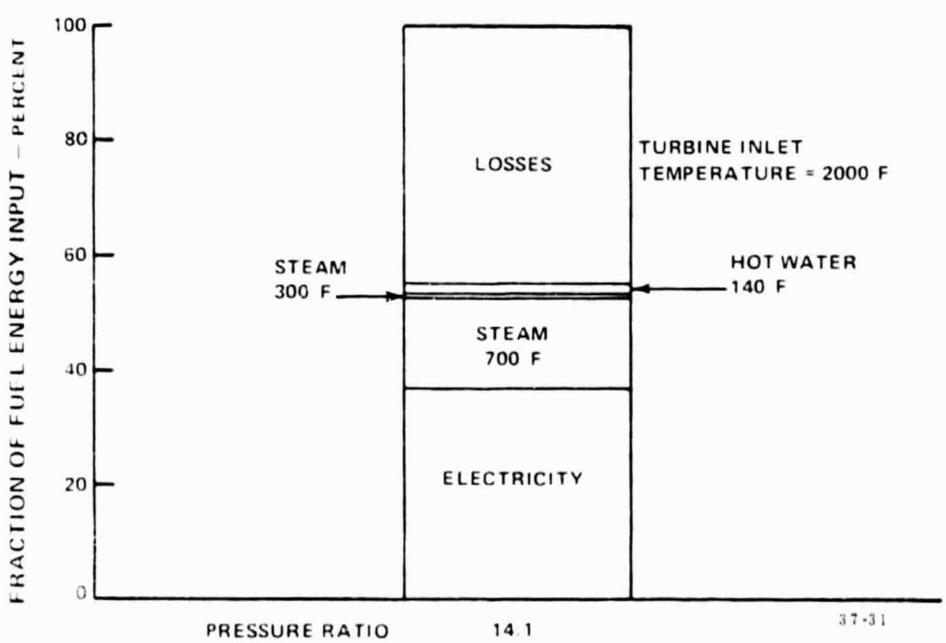


Figure 111-93. Current Gas Turbine Combined Cycle - Distillate Fuel

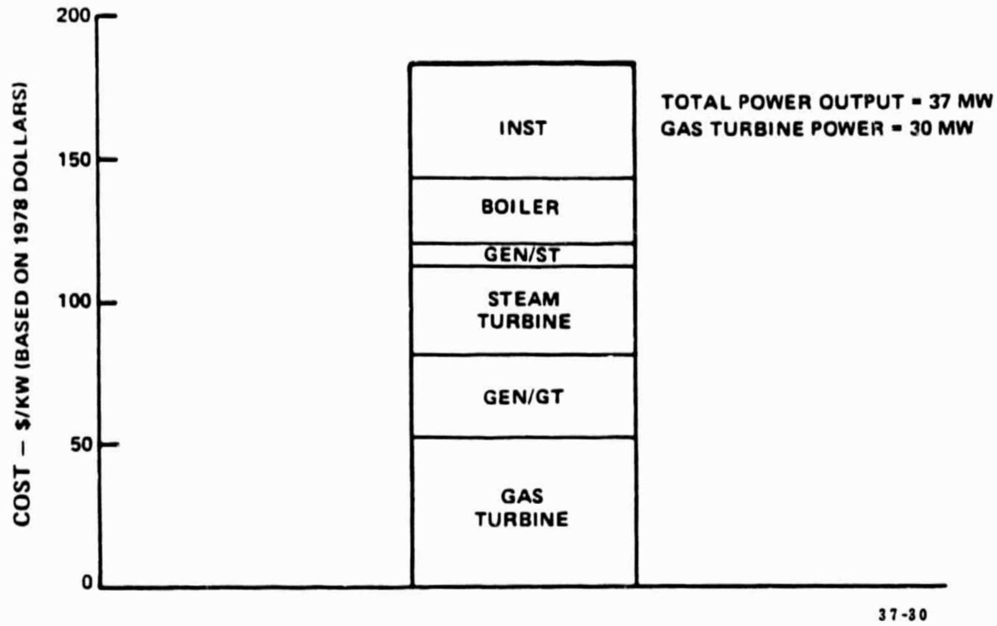


Figure III-94. Current Combined Cycle Estimated Cost

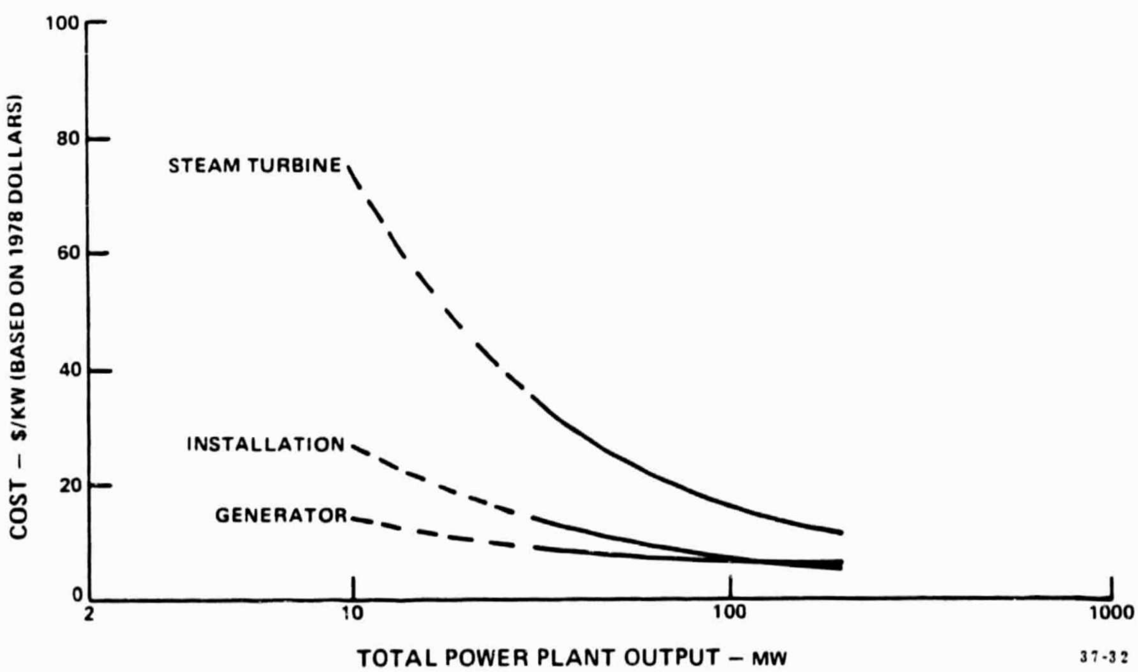


Figure III-95. Steam Turbine - Generator Portion of Current Combined Cycle Estimated Costs

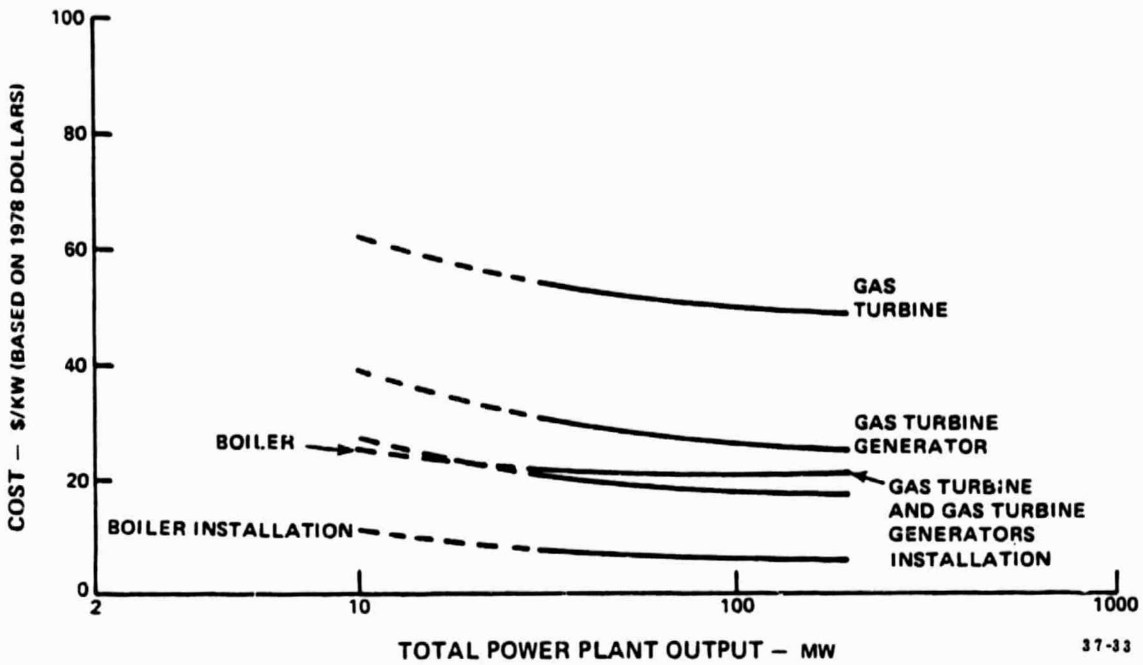


Figure III-96. Gas Turbine - Generator and Boiler Portion of Current Combined Cycle Estimated Costs

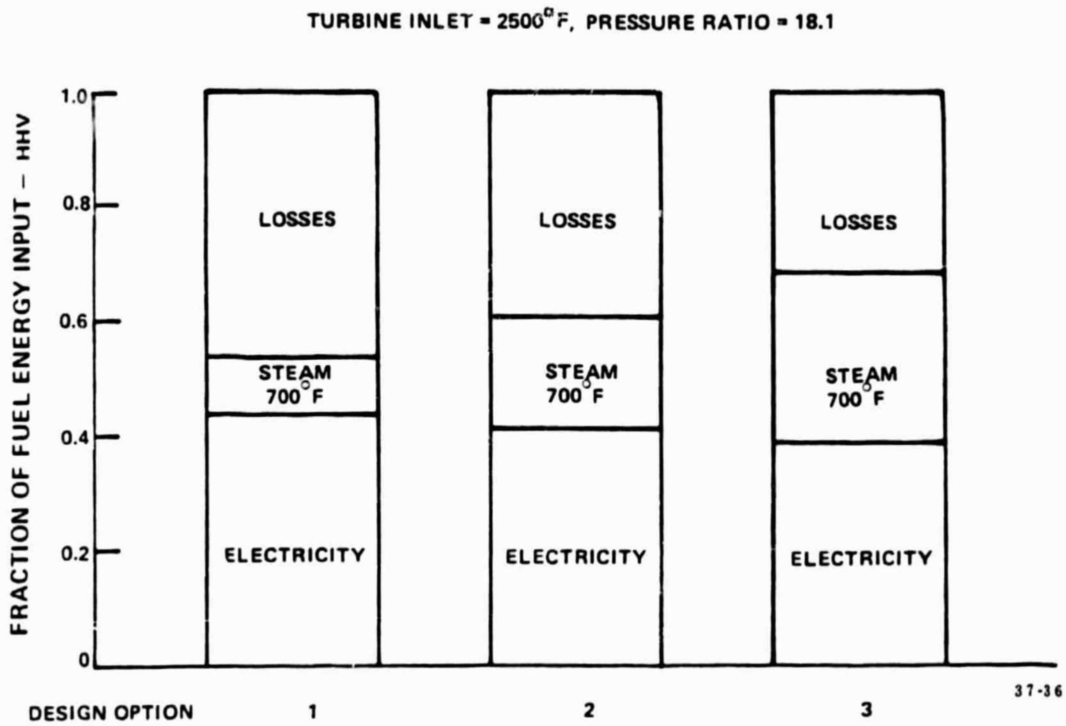


Figure III-97. Advanced Combined Cycle Performance - Petroleum or Coal-Derived Boiler Fuel

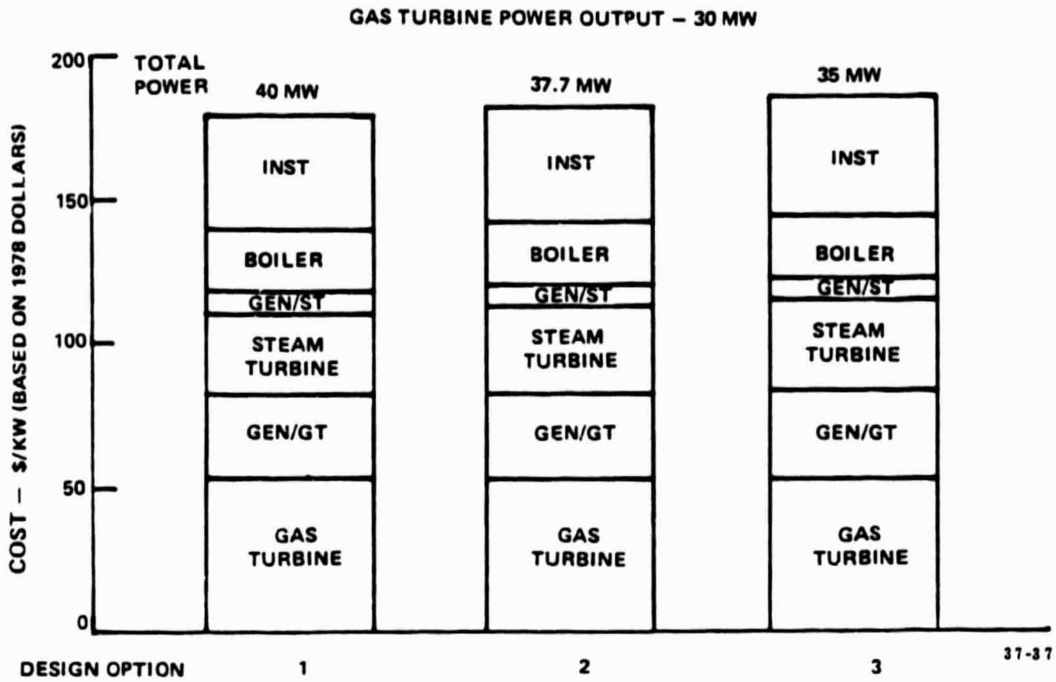


Figure III-98. Advanced Combined Cycle Estimated Cost - Petroleum or Coal-Derived Boiler Fuel

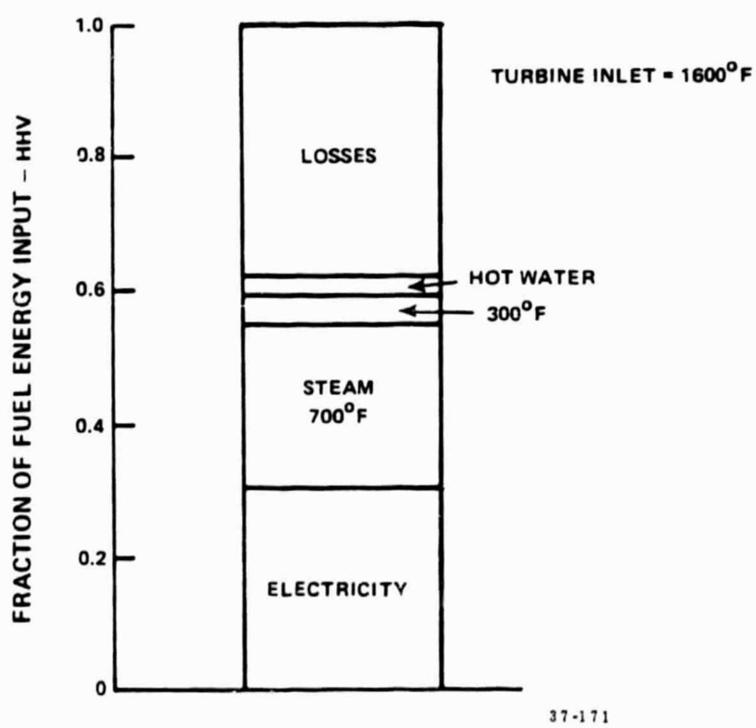


Figure III-99. Advanced Combined Cycle Performance - Direct Coal Fired-Pressurized Fluidized Bed

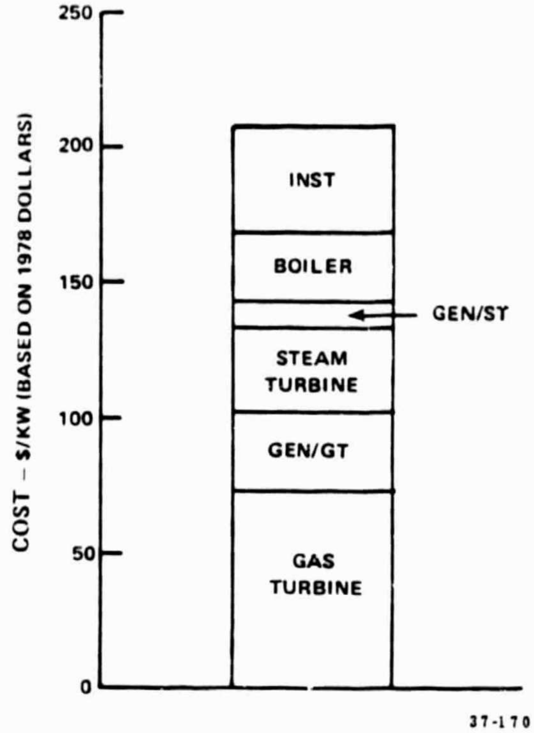


Figure III-100. Advanced Combined Cycle Estimated Cost - Direct Coal Fired-Pressurized Fluidized Bed

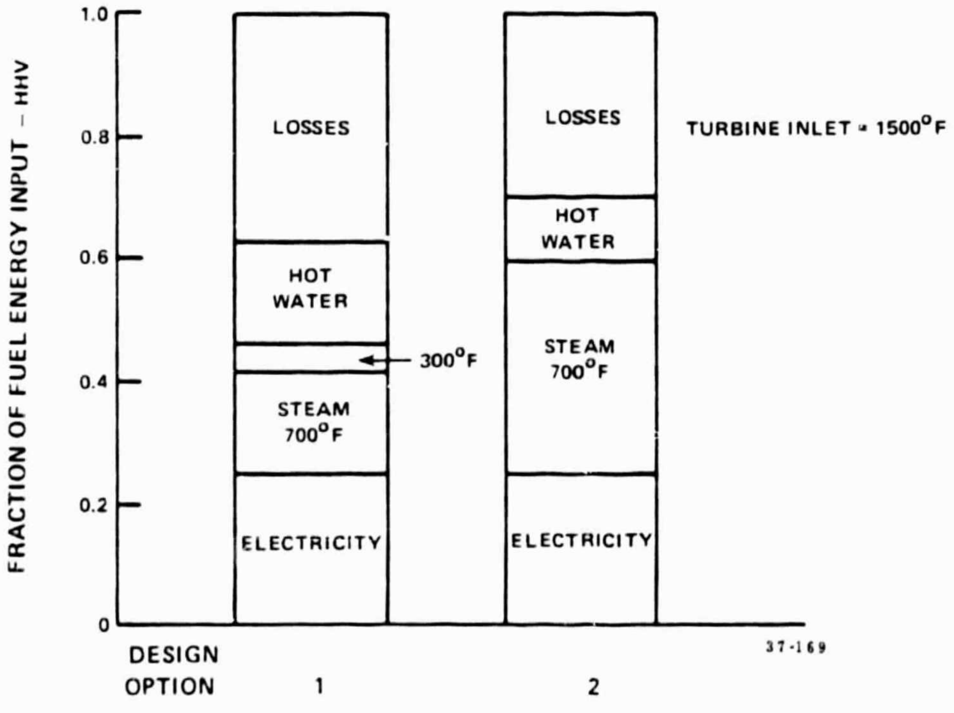


Figure III-101. Advanced Combined Cycle Performance - Indirect Coal Fired-Atmospheric Fluidized Bed

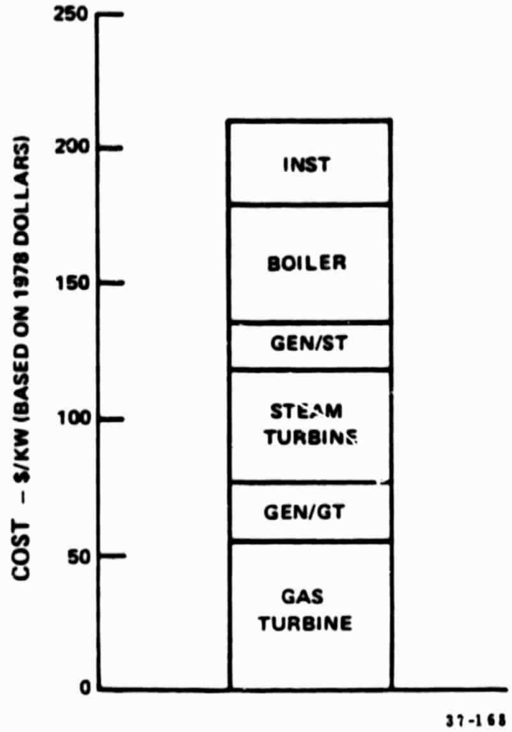


Figure III-102. Advanced Combined Cycle Estimated Costs - Indirect Coal Fired-Atmospheric Fluidized Bed

FUEL CELL POWER PLANTS

INTRODUCTION

The development of fuel cells for utility and on-site applications has been the focus of United Technologies' commercial fuel cell program since the mid 1960's. Efforts under this and other programs have established the technical and operational feasibility of hydrocarbon fuel cells and demonstrated several features which make them attractive. These include high thermal efficiency in small and large sizes as well as the ability to maintain high efficiency at part power. The modular nature of system construction permits factory assembly and convenient transportation to the site. Also, fuel cells emit very low levels of gaseous pollutants and recover water internally such that no external water supply is required. Fuel cells can be located at or near the load.

Many of the above characteristics make the fuel cell attractive as a generator for industrial applications. The potential for heat recovery also makes the concept of cogeneration an attractive option.

Two fuel cell types were included in this study: the low temperature phosphoric acid cell, which has reached the power plant demonstration stage, and the high temperature molten carbonate electrolyte cell which has been demonstrated in Hi-cell assemblies.

An individual fuel cell of either type can be characterized schematically as in Figure III-103 and consists of two electrodes separated by an electrolyte. Fuel is supplied to one electrode, and oxygen (air) is fed to the other. An electric potential slightly below one volt is established. An electric current may be drawn as long as the fuel and air are supplied. Electrodes catalyze the reaction but are not consumed. Single cells can be assembled in series to generate practically any desired voltage. Connecting fuel cell assemblies in parallel permits a variety of power level outputs.

A variety of fuel types were considered in this study including a light petroleum distillate (naphtha), number two oil, a distillate liquid derived from coal, and coal. A fuel processing section is included in the fuel cell power plant to convert hydrocarbon fuels to a hydrogen-rich gaseous mixture more readily utilized by the fuel cell. The cells generate direct current, and therefore an inverter is included to convert this direct current power to conventional alternating current. The combination of the fuel processing section, the fuel cells, the inverter, and associated controls are referred to as a fuel cell power plant.

LOW TEMPERATURE FUEL CELL POWER PLANTS

Current Status

Fuel cells have been in production for space applications for over a decade. They are presently being developed for utility application. In the early 1970's, small 12 kilowatt experimental power plants were installed in 35 locations both in the U.S. and overseas. The purpose of this field test program was to obtain realistic application experience and to establish more comprehensive power plant requirements.

Based on the experience of the 12 kW power plants, a 40 kW unit has been built and is presently under development (see Figure III-104). This power plant incorporates heat recovery, is self sufficient in process water and operates automatically including start up and shut down. A demonstration unit has run in excess of 18,000 hours, has achieved an electrical efficiency of 40 percent, and has demonstrated heat recovery of over 40 percent of the heating value of the fuel.

An experimental one megawatt power plant has been constructed and tested in parallel with electric utility lines. A photograph of this unit is shown in Figure III-105. This pilot plant successfully produced high quality electric power- both real and reactive- to the utility grid, demonstrated rapid transient response, and operated both on light distillate and gaseous fuels.

This experimental unit served as the technical basis for a larger 4.5 MW demonstration power plant presently being constructed in New York City to be operated by Consolidated Edison. The purpose of this test is to confirm operational characteristics, to validate the feasibility of siting in an urban area, and to demonstrate the installation and operation of a fuel cell power plant by a utility. An artist's sketch of this installation is shown in Figure III-106. In the future, power plant modules between 10 - 12 MW are envisioned, with larger size units being built up by operating a number of modules in a parallel arrangement.

Conversion System Description

Two low temperature fuel cell - fuel combinations were included in the study. One (Number 27, Table III-1) is based on current power plant developments and utilitizes a light distillate fuel, such as naphtha, which is converted to a hydrogen-rich process gas utilizing the steam reforming process. The other configuration Number 28 operates on heavier distillate fuel and utilizes an advanced adiabatic reformer concept for fuel processing. Two design options were included for this case.

A flow schematic of the naphtha fueled low temperature fuel cell system (#27) is shown in Figure III-107. Fuel entering the power plant is first desulfurized and then mixed with steam and passed over a catalytic bed in which the reforming process occurs. The product gas from the reformer contains hydrogen along with carbon dioxide, carbon monoxide, and water vapor. This stream is cooled and passed through a catalytic shift converter in which most of the carbon monoxide is converted to hydrogen and carbon dioxide. This gas is then fed to the phosphoric acid fuel cells. Most of the hydrogen is utilized in the cells. The anode discharge stream, consisting of carbon dioxide and some hydrogen is recirculated back to the reformer where the remaining hydrogen is burned to provide the endothermic heat requirements for the reforming reaction. Air is drawn into the power plant and fed to the cells. A small amount of air is consumed in the reformer burner.

The overall process in the fuel cells consists of the continuous electrochemical reaction of hydrogen in the fuel stream and oxygen from air to produce electric power and byproduct water and heat. The byproduct water is removed with the excess air passing through the cells, and this, along with water vapor in the fuel stream exhaust, is recovered in condensers. Sufficient water is recovered to supply the complete process needs of the power plant. Heat is removed from the cells by boiling a portion of circulating water. The steam is separated from the water, and a portion used in the reformer. Regenerative water conditioning equipment is included in the power plant to insure removal of harmful compounds in the steam to the reformer or cell stack water coolant stream.

The system operates at a pressure of 120 psia. Energy available in the reformer burner exhaust is expanded through a turbine which supplies the necessary power for air compression.

The solid state power conditioning subsystem converts the direct current from the fuel cell to regulated alternating current.

Energy conversion system Number 28, design option 1, using coal-derived distillate or number two fuel oil is shown schematically in Figure III-108. The basic operation and component functions are similar to the first system with the exception of the method of fuel processing and sulfur removal. Fuel is mixed with steam and air and gasified in an adiabatic reformer. The reformer utilizes a catalyst to promote the reaction. Unlike steam reforming, the adiabatic reformer generates heat for the reaction in-situ and thus eliminates the need for transferring heat across metal walls. This leads to both lower costs and a simpler reactor vessel.

Sulfur is removed from the fuel prior to entering the shift converter by a regenerable metal oxide scrubbing system. This is a cyclic process in which one bed of absorbent is removing sulfur from the gas stream while the second is being regenerated with air. The sulfur is discharged as sulfur dioxide during regeneration.

Another low temperature fuel cell design option (Number 28, Design Option 2) using coal-derived distillates or number two fuel oil is shown schematically in Figure III-109. The basic operation and component functions are the same as the first design option with coal-derived fuel with the exception of the method of feeding steam to the adiabatic reformer. In the first design option, Figure III-108, product water was first condensed from the fuel cell exhaust gases. Heat from the fuel cell was then used to boil this water for the fuel processor. In the second design option, Figure III-109, fuel cell cathode exhaust, which contains most of the product water in vapor form as the well as unreacted oxygen, is fed directly to the adiabatic reformer. Thus steam is supplied for fuel processing without the intermediate steps of condensation and vaporization. This arrangement provides larger amounts of steam for cogeneration applications, but also results in lower fuel cell electrical output.

The low temperature fuel cell power plant designs described above, use a common cell design. Phosphoric acid electrolyte is contained in a matrix material. The electrodes include platinum catalyst supported on carbon. Operating temperature and pressure are 400°F and 120 psia. These are somewhat higher than present practice of 375°F and 50 psia; however, cells at the higher temperature and pressure are being tested and represent a reasonable extrapolation for future commercial power plants in the 1985-2000 period. Cell power densities are about 200 watts per square foot.

The power plant designs utilize a solid state inverter to provide regulated 60 Hz, 3 phase alternating current. The inverter output voltage can be tailored to a specific industrial requirement from 13.8 kV to 480 volts. The fuel cell power plant can be operated independently or in parallel with other power sources including the electric utility grid.

Performance Characteristics

For each of the low temperature fuel cell power plant designs, a reference design was established with a rated power output of 12 MW. The design point performance of each of these systems is presented in Figures III-110 and III-111 in terms of the fraction of fuel energy available as electrical and thermal output.

The low temperature fuel cell power plant was analyzed for two fuel types, petroleum distillate and coal derived distillate. As can be seen from Figure III-110, the lower quality of the coal distillate (lower carbon to hydrogen ratio) results in a slightly lower steam availability. Figure III-111 illustrates the benefit of utilizing the product water vapor directly from the cells in the fuel processor. The steam availability for industrial use is considerably higher with design option 2 than with option 1 although there is a reduction of electrical output. The amount of 300°F steam is increased and some 500°F steam is produced.

Figure III-112 presents electrical and thermal performance over a range of power plant sizes for the low temperature fuel cell with petroleum distillate fuel. The flat characteristic is indicative of the modular nature of fuel cell systems, which result in the ability to operate efficiently, even in small sizes. The lower limit on power plant size shown in the figures is 400 kW and is consistent with the minimum size requirements of the various industries studied. Characteristics above 12 MW will be constant since larger size power plants would be built up to multiple 12 MW units. The low temperature fuel cells with coal-derived distillate fuel have similar flat performance characteristics over the range from 400 kW to large sizes.

Figure III-113 presents the part power performance for the low temperature fuel cell conversion system number 28, design option 2. Fuel cell power plants provide relatively constant electrical and thermal efficiencies over most of their operating range.

Estimated Costs

Capital cost estimates for the low temperature fuel cell designs were based, in part, on present technology levels and, wherever warranted, on performance projections consistent with experimental activities or analytical models which define methods for performance improvement. In all cases the costs were based on mass production techniques assuming a mature product. Data for costs were based on studies conducted for the government and utilities as well as the experience gained during the construction of the 12, 40 and 1000 kW experimental power plants. Consultants have also been involved in independent estimates for both capital cost and installation costs.

The estimated costs for low temperature fuel cells are presented in Figure III-114. Conversion system number 28 design option 2, which produces the highest quality heat, also results in the highest capital cost. This is due primarily to the poorer performance of the fuel cell caused by a relatively low hydrogen concentration in the anode process gas. The lowest cost fuel cell power plant, conversion system number 27, uses the simplest fuel, naphtha, and the traditional steam reformer for fuel conversion. The variation of low temperature fuel cell power plant costs with power plant rating are presented in Figure III-115. The cost increase at low power is due to the relative increase in the specific cost of such items as controls, structure, and piping.

Installation costs for utility size fuel cell power plants (10 - 30 megawatts) in a mature commercial situation has been estimated for United Technologies by architectural and engineering organizations. The installation cost used in this study of \$10/kilowatt was based on these prior estimates with the balance of plant items removed.

The operating and maintenance costs anticipated for low temperature fuel cell power plants were estimated based on a 30 year installation lifetime. Operation was assumed to be virtually continuous with the power plant operating 8000 hours per year at rated power. Scheduled replacement of major components was included

in the maintenance costs. The fuel cell assemblies were assumed to be replaced after 40,000 hours of operation. In addition, routine maintenance schedules were established, and labor estimates made to accomplish these tasks. The resulting operating and maintenance costs for the low temperature fuel cell systems are shown in Table III-42. The major difference between the systems using naphtha, conversion system number 27, and coal-derived distillate, conversion system number 28, is the replacement costs associated with the fuel processing catalyst and sulfur removal absorbent.

Emissions

The estimated emissions from the low temperature fuel cell power plants, Table III-43, are based on the measured emissions from experimental power plants, Table III-44. All systems studied fall well below the emissions guidelines provided by NASA.

For the coal-derived fuel, conversion system no. 28, the specification indicates a sulfur level of 0.5 weight percent. Technically, absorption of the sulfur (as hydrogen sulfide) on disposable zinc oxide could be employed. However, the associated logistics of zinc oxide supply and zinc sulfide replacement suggested a regenerable system. A metal oxide desulfurizer was selected for this design. The sulfur in the gas stream forms metal sulfide and is retained. With two metal oxide units, one absorbs sulfur while the other is separated from the gas stream, exposed to air, the metal oxide is regenerated, and sulfur is released as sulfur dioxide. For the sulfur content of the fuels specified, 0.57 lbs of sulfur dioxide is released per million Btu of fuel input. If pollution restraints at the industrial site are severe, the metal oxide could be transported to an acceptable location for disposal or regeneration.

For the petroleum distillate, conversion system number 27, the sulfur level in the fuel is at least an order of magnitude lower than for the coal-derived product. It is therefore quite feasible to use disposable zinc oxide, and this would result in essentially zero emission of gaseous sulfur components at the fuel cell site.

The nitrogen oxide emissions (NOx) are essentially an order of magnitude lower than the guidelines for both coal-derived and natural fuels. This is due to the fact that a large fraction of the fuel is consumed in the electrochemical process. The remainder of the fuel, a mixture of hydrogen and carbon dioxide, is burned, but at relatively low flame temperatures, which results in low production of thermal NO_x.

Petroleum distillate (naphtha) fuel contains essentially no particulate matter. The coal-derived fuel contains up to 0.06 weight percent ash. For this study, it was assumed that all of the ash would leave the power plant as particulate matter. This amounts to 0.03 lbs per million Btu of fuel input, or about one third the amount permitted by the guidelines.

Noise emissions by the power plant are caused primarily by the rotating equipment, the turbocompressor. Estimates of noise levels measured 100 feet from the power plant boundary are 55 db (A).

Physical Characteristics

The low temperature fuel cell power plants included in this study physically require about 1.5 square feet and 24 cubic feet per kilowatt. The installation period is estimated to be about one year.

Cogeneration Applicability

The low temperature fuel cell power plant embodies the principal characteristics necessary for successful cogeneration applications. Specifically, the power plant offers siting and operating flexibility. The low level of pollutants and the low noise may be particularly important in many industrial locations. The fuel cell power plant is able to operate efficiently over a wide range of output levels and is capable of very rapid response to variations in demand. These units are suitable

for either grid connected or independent electrical operation. Since the fuel cell power plants are modular, additions to meet expanded plant requirements are straight forward. Since low temperature fuel cells typically operate at 400°F, the capability to raise high pressure steam is limited. However, power plant configurations emphasize the production of 300°F and 500°F steam.

The low temperature fuel cell power plant can operate with a variety of fuels in the 1985 - 2000 time period. Pipeline gas and naphtha are well established as fuel cell power plant fuels. No. 2 fuel oil and coal-derived distillate fuel capability are being developed. With the study guidelines on-site coal gasification represents a possible fuel option for fuel cell power plants. For this study the on-site coal gasification possibility was explored with the high temperature fuel cell. Since the result was encouraging, the on-site coal gasification could well be a viable option for the low temperature fuel cell as well.

Future Developments

Low temperature fuel cell power plants have demonstrated the technical capabilities required for cogeneration applications. However, to achieve widespread cogeneration use in the 1985 - 2000 period, the following technical developments are required.

1. Engineering development to achieve required levels of capital cost and durability to provide commercially competitive equipment.

2. Development of fuel processing and desulfurization equipment, including the adiabatic reformer, to full scale commercially viable subsystems for coal derived distillate fuels.

3. Development of the cells to provide improved performance and durability consistent with higher pressure and higher temperature operating conditions.

4. Development of heat recovery equipment and controls.

5. Development of recycle systems to provide high temperature heat recovery.

HIGH TEMPERATURE FUEL CELL POWER PLANTS

Current Status

High performance fuel cells operating at elevated temperature and employing molten carbonate electrolyte are being investigated. This cell technology has been under investigation in both Europe and the United States since the 1950's. It has attractive features in terms of both improved performance and the elimination of noble metal electrode catalysts. Cell development, however, has been limited by materials able to withstand the operating temperature and corrosive electrolyte for long periods of time.

Present cell configurations operate at about 1200°F and utilize electrodes constructed of porous nickel. The electrolyte is a mixture of alkali metal carbonates supported in a ceramic tile. The electrolyte is solid at room temperature but becomes liquid at operating temperature.

Present research and development activities are concentrating on cell and cell assembly (stack) development as a necessary precursor to power plant construction or testing. Most of the experimental cells have been investigated at atmospheric pressure. Recently higher pressure cells (up to 120 psia) have been tested. A steady performance improvement has been accomplished over the past 10 years.

Molten carbonate fuel cells have been built in practical sizes up to an active area of about one square foot. A 19 cell assembly has operated for over 1000 hours.

The longest individual cell test is 43,000 hours. This cell demonstrated the potential of achieving the endurance goal of 40,000 hours. Recently, a cell utilizing an advanced design configuration and advanced materials was tested for over 15,000 hours.

Conversion System Description

Three high temperature fuel cell power plant - fuel combinations were included in the study:

Conversion System Number	Molten Carbonate Fuel Cell P/P Fuel
No. 29	Petroleum distillate (No. 2 fuel oil)
No. 30	Coal-derived distillate
No. 31	Coal with on-site gasifier

The system utilizing coal directly employed an air-blown Texaco gasifier with a physical absorption process for sulfur removal.

A flow schematic for conversion systems 29 and 30 is shown in Figure III-116. Fuel along with superheated steam and air are fed to a adiabatic reformer where a hydrogen-rich gas is produced. This gas, after giving up a portion of its sensible heat to produce steam for industrial use, is then desulfurized and fed to the fuel cell anodes. A large fraction of the hydrogen is consumed by the cells. The fuel stream leaving the cell stacks passes through a condenser which provides process water. The remaining gas leaving the condenser is then burned and, along with process air, fed to the fuel cell cathodes. At the cathode, both oxygen and carbon dioxide are utilized in the electrochemical process. The exhaust is expanded through a turbine; the resulting shaft power is used to compress air for both the adiabatic reformer and the fuel cells. The system is operated at a nominal pressure of 120 psia. A shift converter for carbon monoxide conversion is not required as in low temperature fuel cell systems. A portion of the product steam is used in the reformer, and the remaining steam is available for cogeneration. Water recovery is sufficient and an external water supply is not required.

As in the previous systems, a solid state inverter is used to convert the direct current to conventional regulated 60 Hz, 3 phase alternating current.

Energy Conversion System Number 30, design option 3, emphasizes steam production for cogeneration. A portion of the anode exhaust gas (which contains the water produced by the electrochemical reaction) is fed directly to the adiabatic reformer. Therefore, the steam for the reformer does not have to pass through a condenser and be subsequently boiled. This arrangement results in larger amounts of high quality steam available for cogeneration applications but require a somewhat larger reformer and fuel cell assembly.

Energy conversion system number 31 utilizes coal for fuel and is shown in Figure III-117. The system employs an entrained flow, air-blown gasification plant operating at 600 psig which converts coal into a stream containing essentially hydrogen, carbon oxide and nitrogen. The gasifier effluent, which is at approximately 2400°F, is cooled by generating steam for cogeneration. In addition, particulate matter such as ash and soot, is removed in a water wash. The cooled gas is then passed through a Selexol desulfurization unit where almost all of the sulfur bearing gases, such as hydrogen sulfide (H_2S) and carbonile sulfide (COS) are removed to a level below 1 PPM. The Selexol plant is regenerable, and the sulfur compounds are finally converted to elemental sulfur in a Claus plant. The clean fuel gas, leaving the desulfurizer, is expanded through a power recovery turbine (not shown in Figure III-117) to 150 psia, which is the operating pressure of the fuel cells. The fuel cell heat is used to raise high quality steam for cogeneration. Cell exhaust gases are expanded through a turbine which drives the compressor supplying process air to the gasifier and to the fuel cells.

Performance Characteristics

For each of the systems described above a reference design was established. For the liquid fueled power plants, 12 MW was selected. For the system operating on

coal, the reference size was 100 MW. Various design options were employed for each of the basic systems. Two options were provided for energy conversion system no. 29 with petroleum distillate fuel. One design was aimed at high electrical efficiency and the second emphasized steam generation. Two corresponding designs were defined for energy conversion system no. 30, which consumes coal derived distillate fuels. The performance for these four designs is shown in Figure III-118. The amount of steam produced for industrial processes is limited. A third design option for conversion system no. 30 employed the alternate configuration where a portion of the anode exhaust stream containing water vapor is fed directly to the reformer. The performance for this design option is shown in Figure III-119 along with the two previous options. This option provided the greatest amount of steam for cogeneration. A fourth design option was included for conversion system no. 30 to enhance applicability in a wide range of cogeneration situations. Since molten carbonate fuel cells operate at 1200F, higher temperature heat could be available. As indicated in Figure III-119, the fourth option provided a clean, hot gas stream at 1000F at the expense of 700F steam.

Energy conversion system no. 31, which utilizes the coal directly, results in a relatively low electrical efficiency, 27.6 percent, because of the losses associated with coal gasification. However, a significant quantity (34.4 percent) of the fuel energy is available as 700F steam. The design point performance for this power plant is presented in Figure III-120.

Figure III-121 presents electrical and thermal performance over a range of power plant sizes for energy conversion system no. 30, design option no. 2. The flat characteristic is indicative of the modular nature of fuel cell systems which result in the ability to operate efficiently even in small sizes. The lower limit on power plant size indicated in Figure III-121 is 400 kilowatts, which is consistent with the smallest size pressurized system for utility and industrial application. Characteristics above 12 megawatts are constant since larger sized power plants are built up

of multiple 12 megawatt units. Energy conversion system no. 31 which uses coal directly, was extrapolated down to 5 megawatts in size. Specific costs are expected to increase significantly for smaller size coal consuming systems.

Figure III-122 presents the part power performance for energy conversion system no. 30 design option no. 3. A similar trend is typical for most systems.

Estimated Cost

Capital cost estimates were made for the three high temperature fuel cell energy conversion systems, numbers 29, 30, and 31. These estimates utilize data developed in the low temperature fuel cell activity as appropriate. However, since there has been much less experimental power plant experience with the high temperature fuel cell, these estimates are of necessity more approximate. They are based, to a larger extent, on technology extrapolations and on material and fabrication cost estimates which do not have multiple, detail analyses.

The results of the cost studies for the three systems at the rated power are presented in Figure III-123. The coal gasification system is the most expensive, due for the most part to the high cost of the gasifier and sulfur cleanup.

The variation of the power plant costs as a function of rating is presented in Figure III-124. The liquid fueled systems have relatively flat specific cost characteristics, while the option with a coal gasifier becomes significantly more expensive in the smaller sizes.

The predicted operating and maintenance costs for the three high temperature fuel cell power plant configurations are shown in Table III-45. The ground rules for these estimates are essentially the same as described for the low temperature fuel cell system. The operations and maintenance costs for the system utilizing coal directly include the gasifier. Costs are included for replacement of chemicals and catalysts for the desulfurization and sulfur recovery systems as well as maintenance for the gasifier vessels and steam generators.

Emissions

Table III-46 summarizes the emission characteristics for the three high temperature fuel cell cases studied. The sulfur level and particulate emissions for the coal-derived liquid cases are similar to those for the low temperature cells. The nitrogen oxides, however, are slightly higher for the molten carbonate cases since ammonia formed in the fuel processing step is assumed to burn and produce nitrogen oxide.

For energy conversion system number 31 which utilizes coal directly, the gaseous sulfur emissions are very low since almost all the sulfur in the coal is scrubbed out of the process gas and converted to elemental sulfur. Only the SO_2 emitted from the sulfur recovery system is released to the atmosphere, and this is estimated to be no greater than 0.07 lbs per 10^6 Btu.

The nitrogen oxide level for the coal case is higher than for the liquid fueled cases but still falls well below the guidelines. The higher nitrogen oxide level is caused by the high amounts of fixed nitrogen in the coal. Although some of this nitrogen is converted to ammonia, which is washed from the process gas, the residual ammonia in the gas is assumed to be burned and to produce nitrogen oxide directly.

The particulate matter leaving in the exhaust of the coal consuming plant is very small primarily due to the water wash and cleansing action of the sulfur clean-up system.

Physical Requirements

The liquid fuel, high temperature fuel cell power plants included in this study physically require about 1.5 square feet per kilowatt. Installation period is estimated to be about one year. The system with the coal gasifier will typically need 2.5 square feet per kilowatt and require approximately 3 years for installation.

Cogeneration Applicability

The molten carbonate cell offers promise for second generation fuel cell power plants in terms of achieving higher electrical efficiency and higher quality heat for cogeneration applications. These power plants will embody the same characteristics as the low temperature fuel cells which make them appropriate for cogeneration: siting flexibilty, low on-site emissions, operating versatility, and ease of expansion.

These high temperature fuel cell power plants can also operate on a variety of fuels in the 1985 - 2000 time period. Pipeline gas and naphtha, no. 2 oil, and coal-derived distillate fuel capabilities are being developed. On-site coal gasification represents a possible option particularly for the larger installations.

Future Developments

To achieve widespread cogeneration use in the 1985 - 2000 period, the following technical developments are required.

1. Development of materials and configurations to achieve the required durability and capital cost of fuel cell assemblies.

2. Development of fuel processing and desulfurization techniques for coal-derived distillate type fuel to full scale commercially viable subsystems.

3. Power plant design and development of the complete system and the various components within the system.

4. Development of small scale coal gasification units with desulfurization capabilities suitable for fuel cell power plants.

TABLE III-42. LOW TEMPERATURE FUEL CELL OPERATING AND MAINTENANCE COST ESTIMATES

Conversion System		Mils per Kilowatt-Hour
27	Petroleum Distillate - Naphtha	2.2
28	Coal-Derived Distillate - #2	2.9

TABLE III-43. PREDICTED LOW TEMPERATURE FUEL CELL POWER PLANT EMISSIONS

	Pounds per Million Btu Fuel		
Conversion System	27 Petroleum Distillate Naphtha	28 Coal-Derived Distillate Option 1	28 Coal-Derived Distillate Option 2
Sulfur Dioxide	NIL	0.57	0.57
Nitrogen Oxides	0.016	0.125	0.042
Particulates	NIL	0.034	0.034

TABLE III-4. MEASURED FUEL CELL POWER PLANT EMISSIONS WITH NATURAL GAS AND NAPHTHA FUELS

	Pounds per Million Btu Fuel
Power Plant Size:	12 kW - 1 MW
Sulfur Dioxide	0 - 0.000003
Nitrogen Oxides	0.014 - 0.024
Particulates	0 - 0.00003

TABLE III-45. HIGH TEMPERATURE FUEL CELL OPERATING AND MAINTENANCE COST ESTIMATES

Energy Conversion System		Mils per Kilowatt-Hour
29	Petroleum Distillate Fuel	2.7
30	Coal-Derived Distillate Fuel	2.3
31	Coal	3.0

TABLE III-46. PREDICTED HIGH TEMPERATURE FUEL CELL POWER PLANT EMISSIONS

Energy Conversion System	29	30	31
Emissions - Pounds per Million Btu Fuel Input			
Sulfur Dioxide	0.51	0.57	0.07
Nitrogen	0.083	0.087	0.201
Particulates	NIL	0.034	NIL

References

Final Report, EPRIAF-753, Project 239.

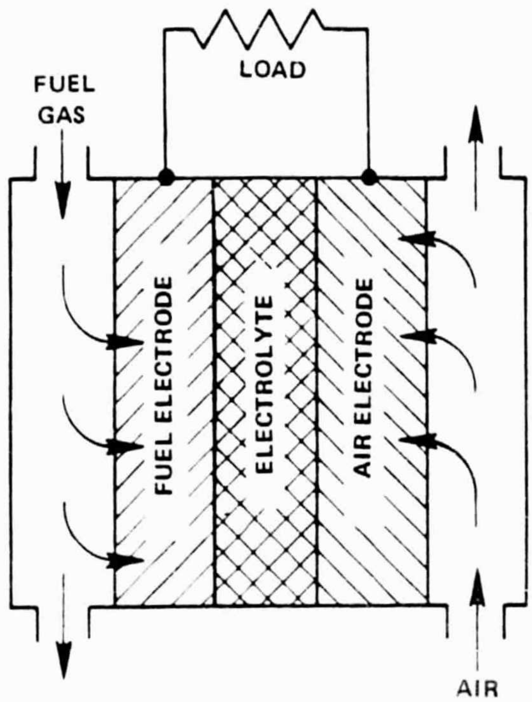
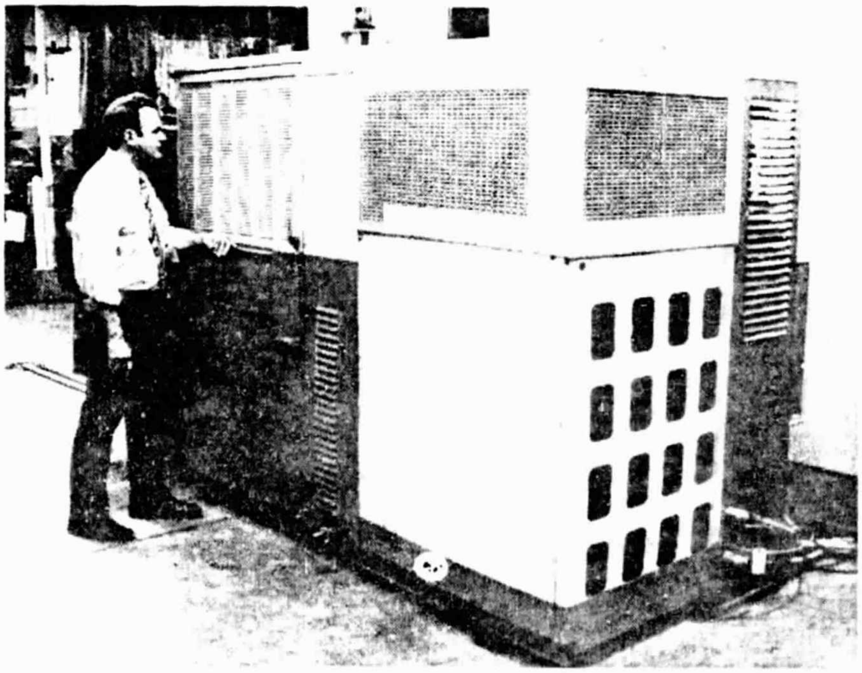
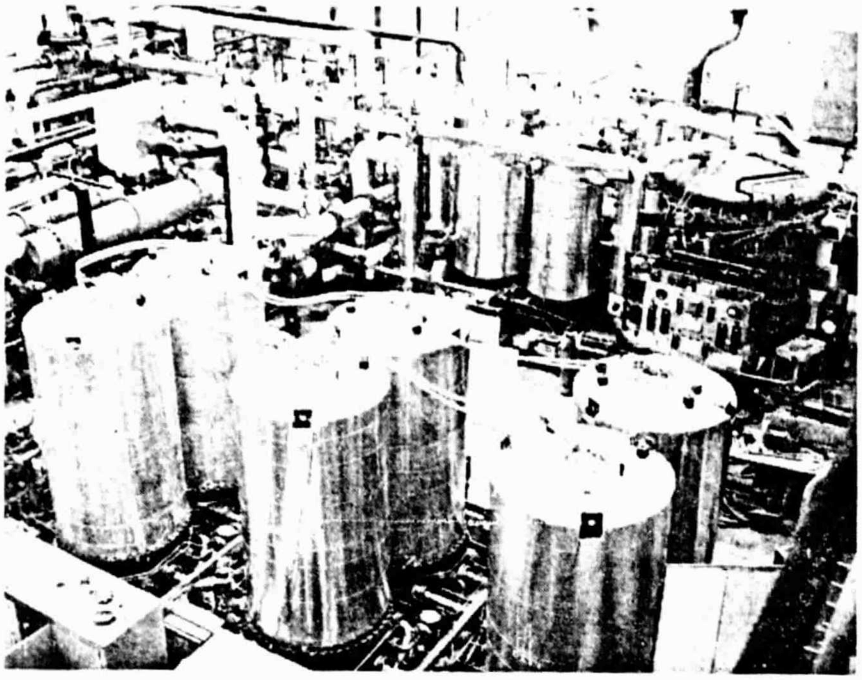


Figure III-103. Fuel Cell Schematic



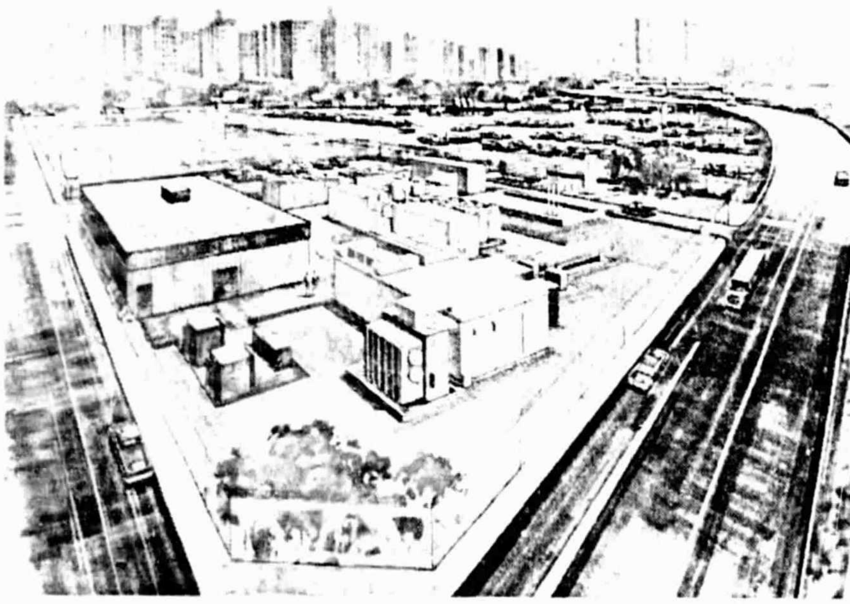
(WCN-3608)

Figure III-104. 40 kW Fuel Cell Power Plant



(WCN-4623)

Figure III-105. 1 MW Pilot Fuel Cell Power Plant



(WCN-5454)

Figure III-106. 4.5 MW Utility Fuel Cell Demonstration

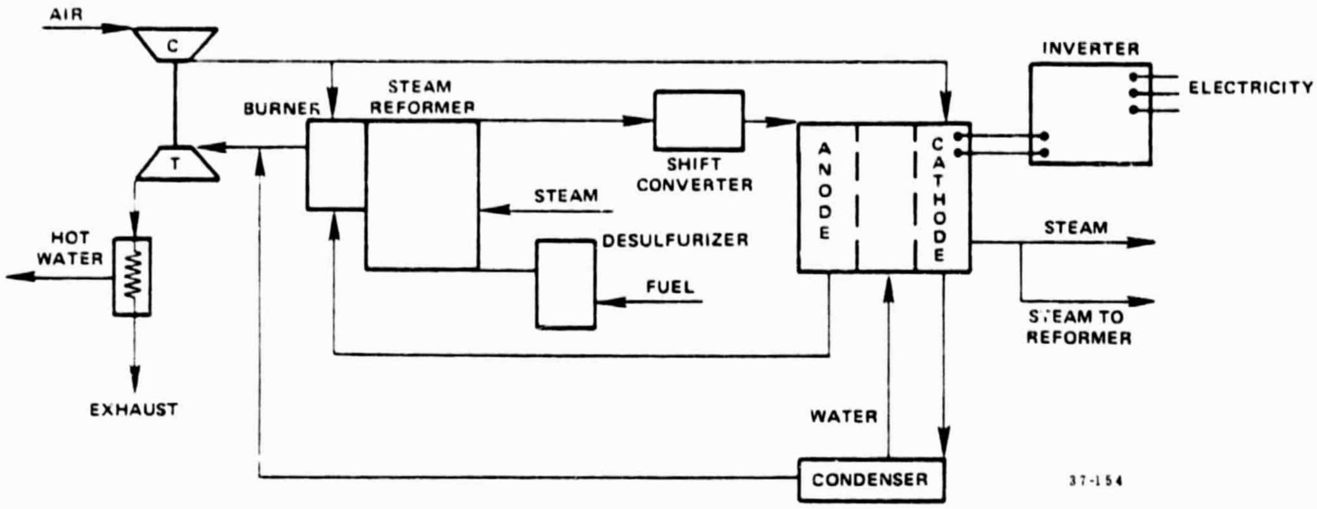


Figure III-107. Low Temperature Fuel Cell Power Plant Schematic - Petroleum Distillate (Naphtha) Fuel (Conversion System #27)

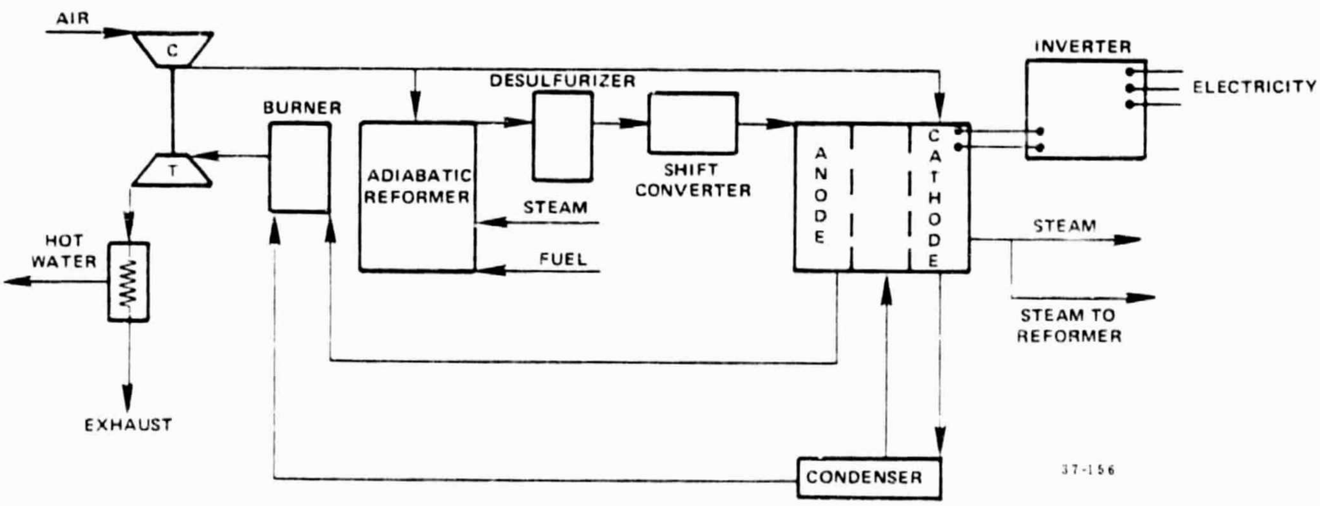


Figure III-108. Low Temperature Fuel Cell Power Plant Schematic - Coal-Derived Distillate Fuel (Conversion System #28, Design #1)

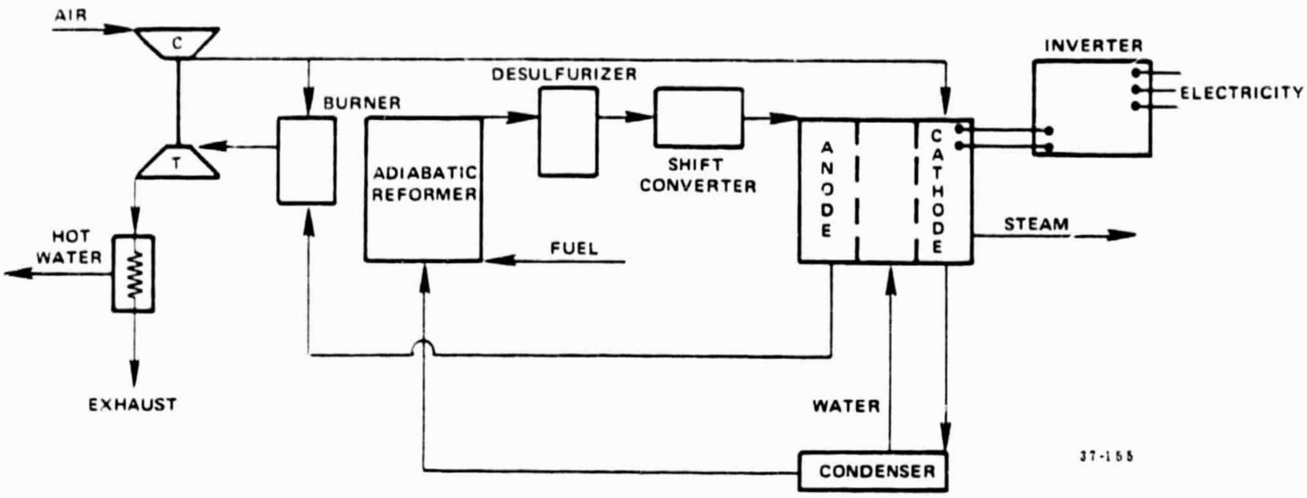


Figure III-109. Low Temperature Fuel Cell Power Plant Schematic - Coal-Derived Distillate Fuel (Conversion System #28; Design #2)

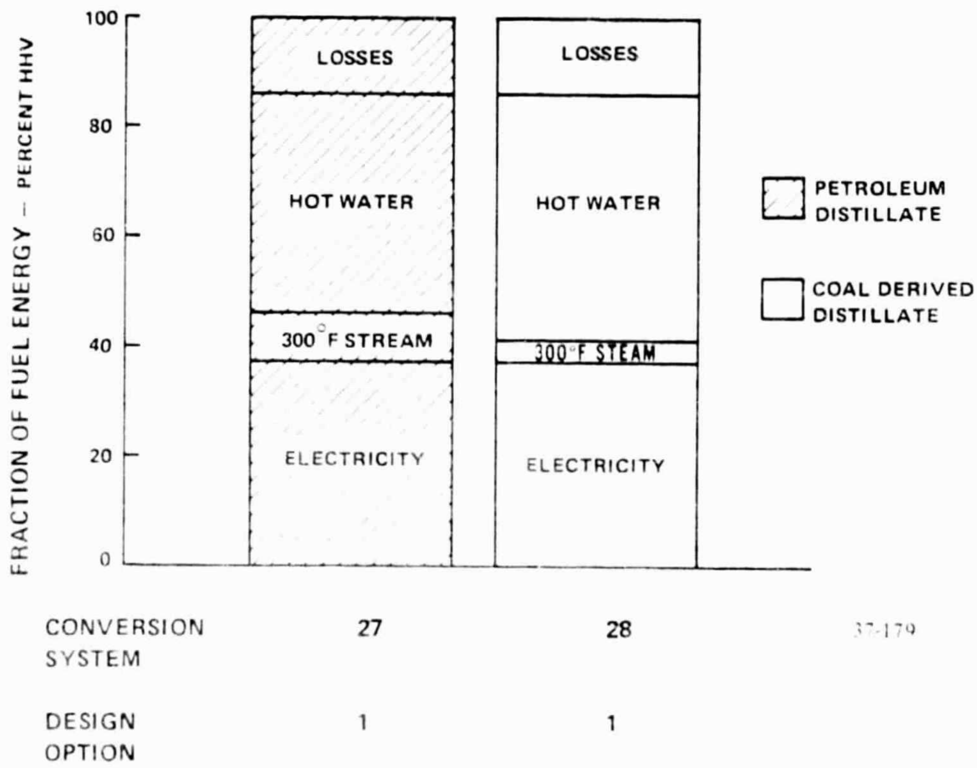


Figure III-110. Low Temperature Fuel Cell Performance - Petroleum and Coal-Derived Distillate Fuel

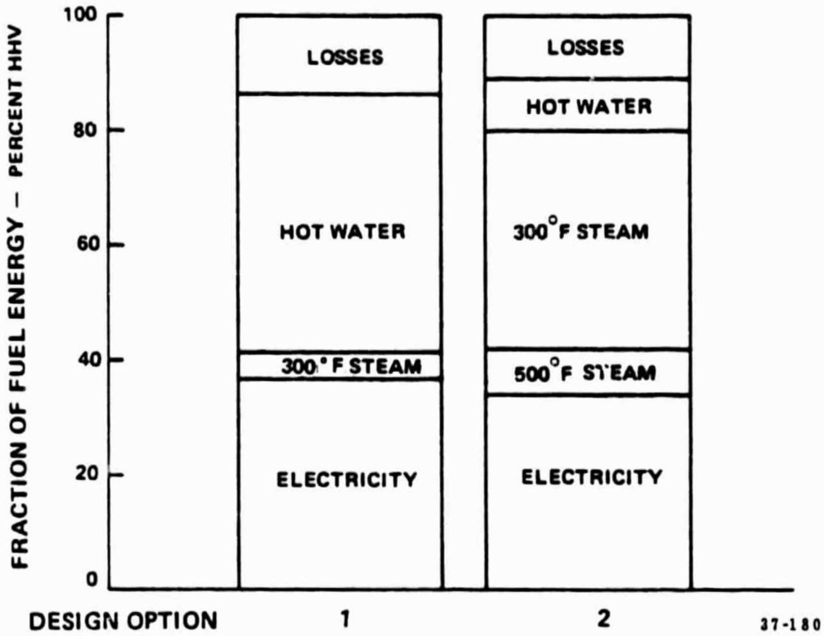


Figure III-111. Low Temperature Fuel Cell Number 28 Performance - Coal-Derived Distillate Fuel

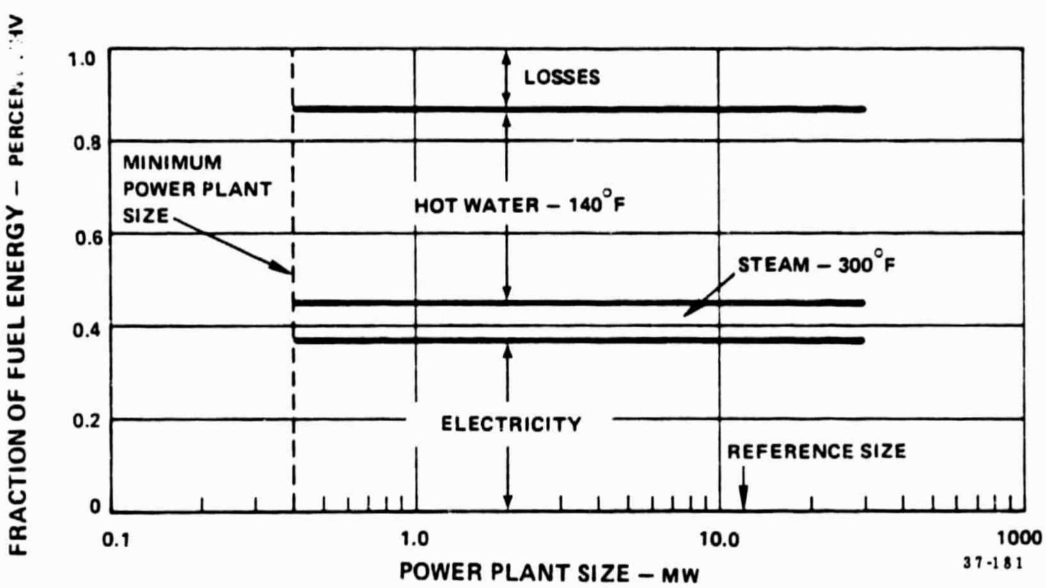


Figure III-112. Low Temperature Fuel Cell Performance at Various Sizes - Petroleum Distillate (Naphtha Fuel)

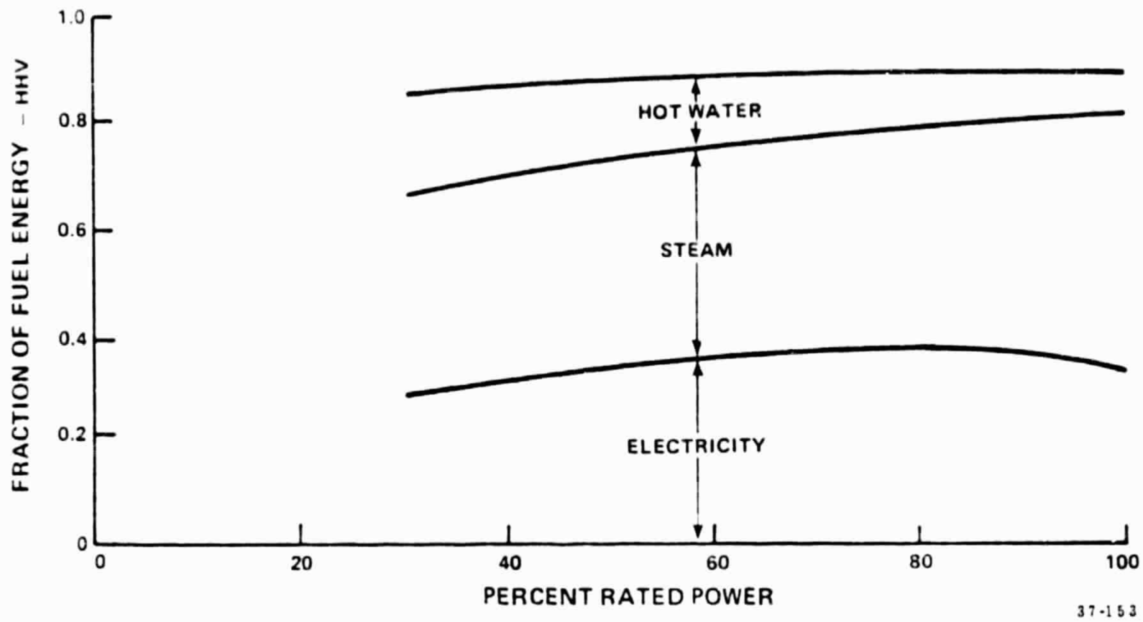


Figure III-113. Off-Design Performance of Low Temperature Fuel Cell Power Plant - Coal-Derived Distillate Fuel - Design Option #2

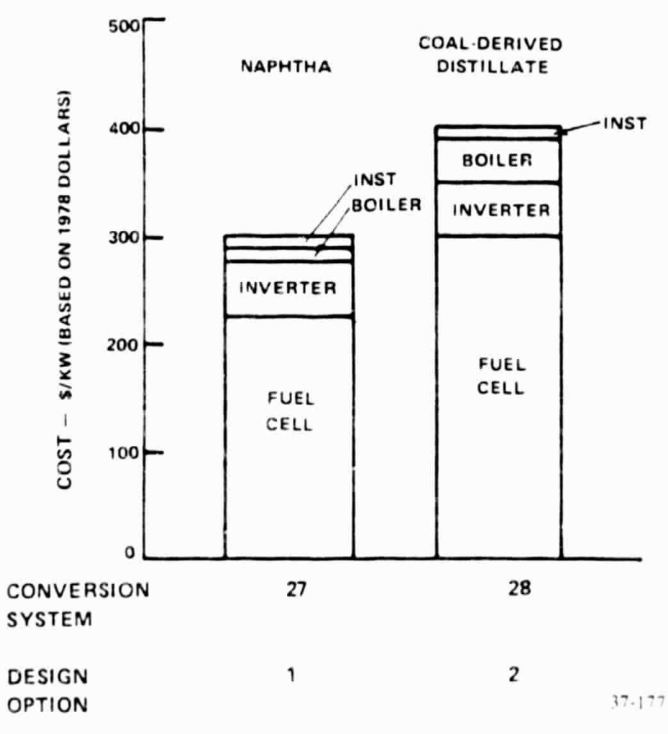


Figure III-114. Low Temperature Fuel Cell Estimated Costs

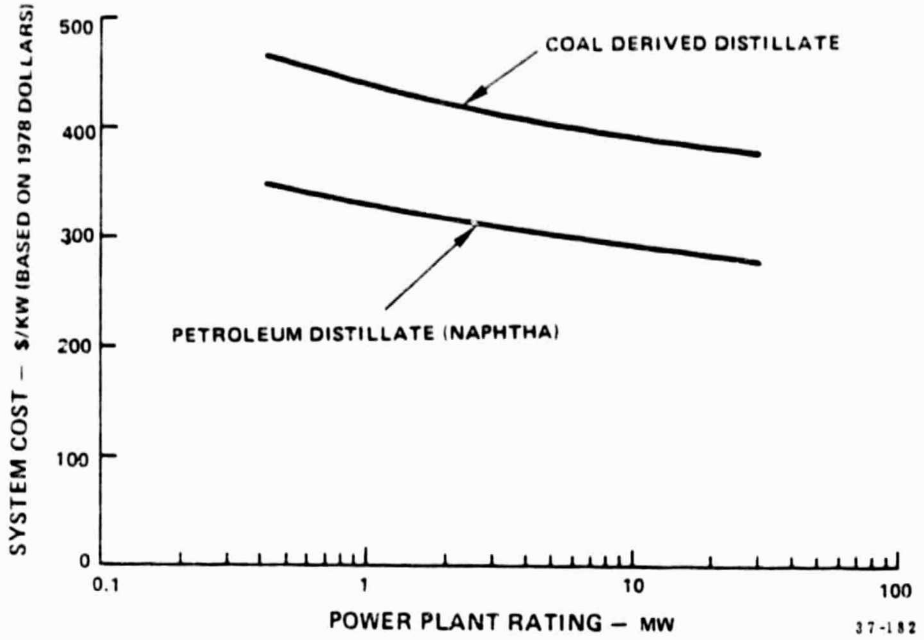


Figure III-115. Low Temperature Fuel Cell Estimated Cost Variation with Size

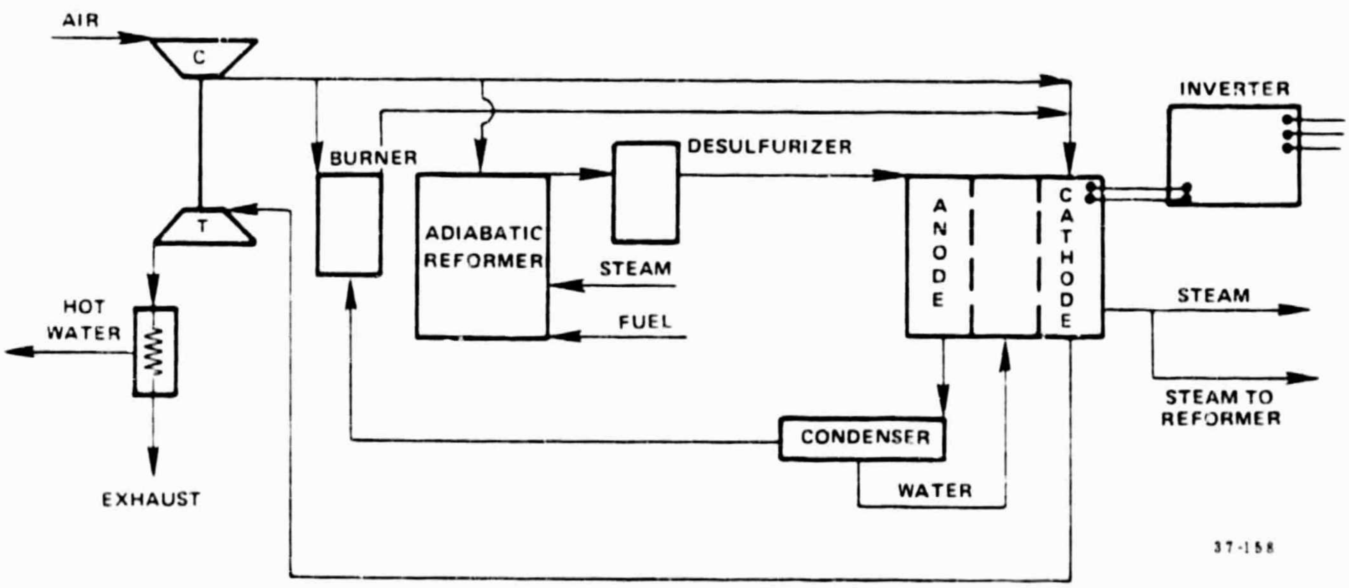


Figure III-116. High Temperature Fuel Cell Power Plant Schematic - Petroleum or Coal-Derived Distillate Fuel - Conversion Systems 29 and 30

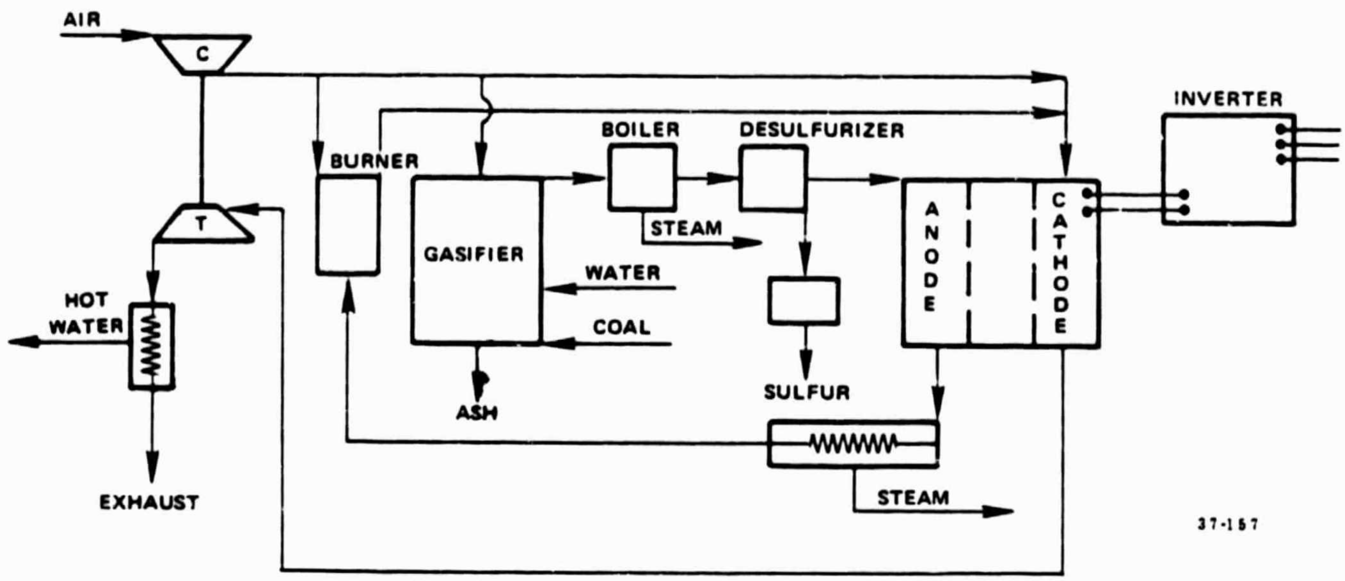


Figure III-117. High Temperature Fuel Cell Power Plant Schematic - Gasified Coal Fuel - Conversion System 31

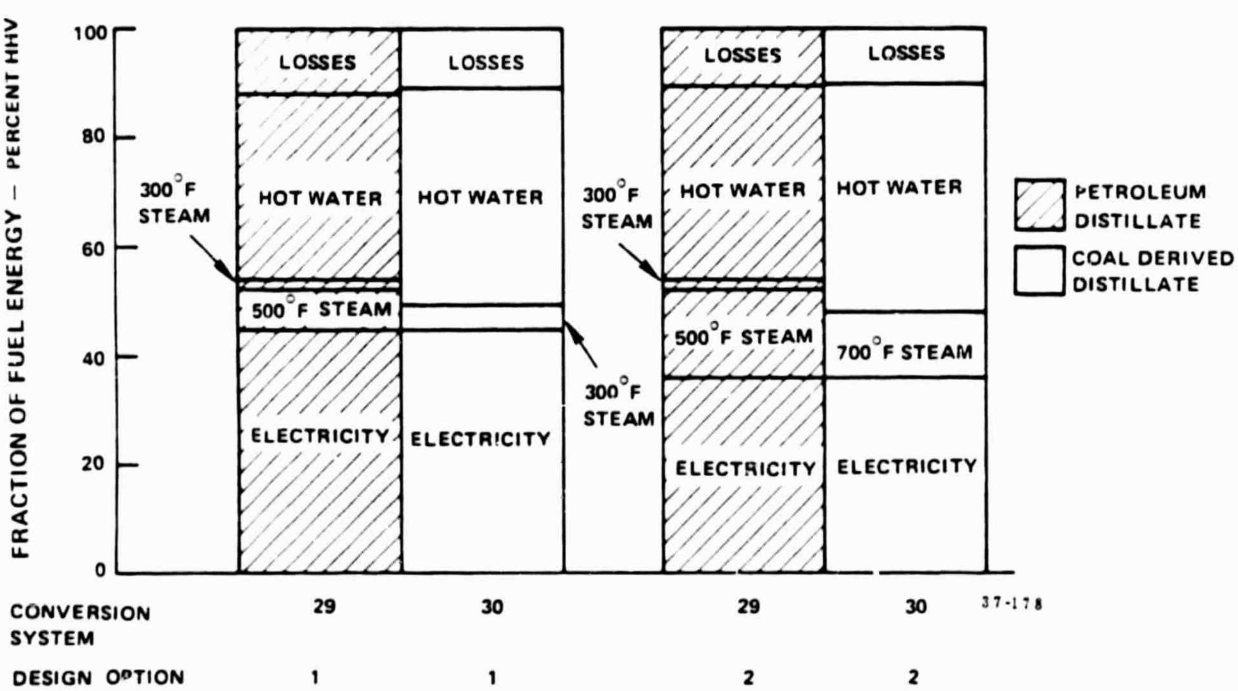


Figure III-118. High Temperature Fuel Cell Power Plant Performance - Distillate Fuel

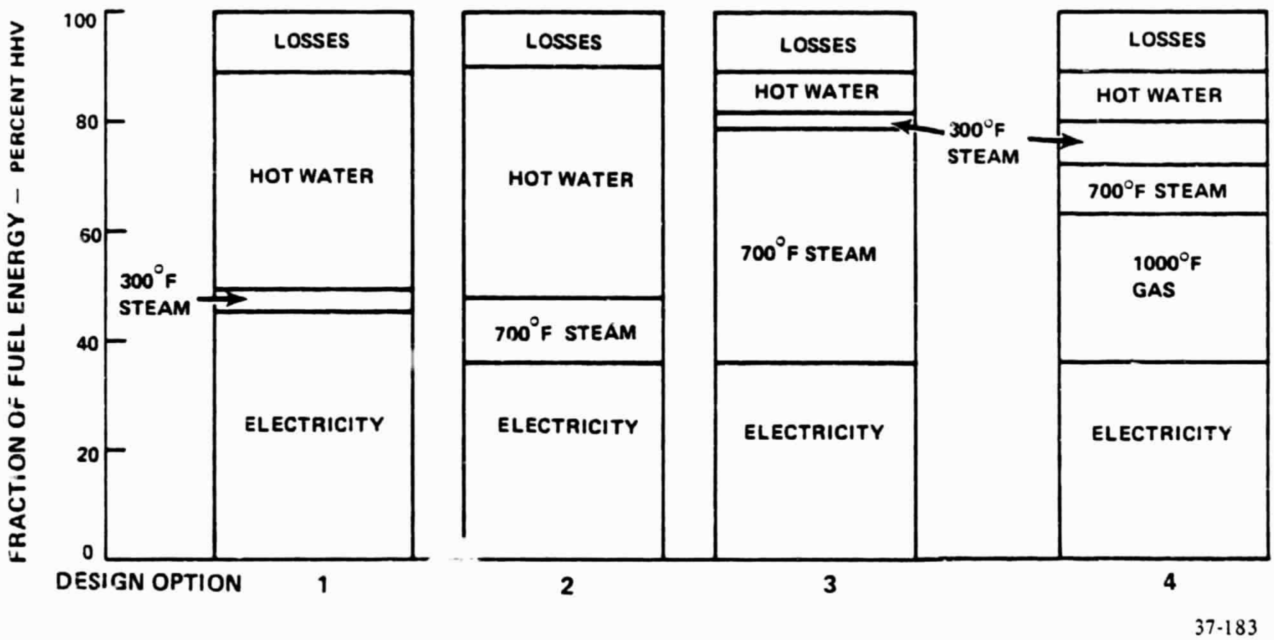


Figure III-119. High Temperature Fuel Cell Power Plant Performance - Coal-Derived Distillate Fuel - Conversion System 30

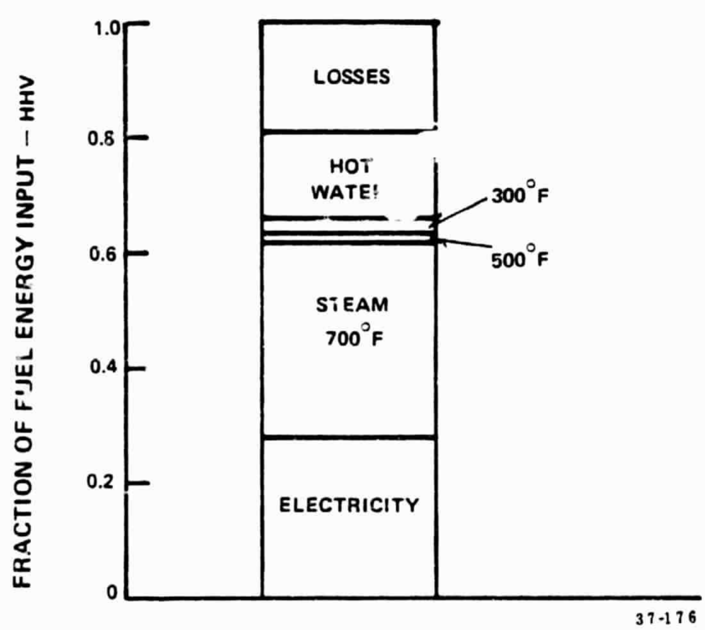


Figure III-120. High Temperature Fuel Cell Power Plant Performance - Gasified Coal Fuel - Conversion System 31

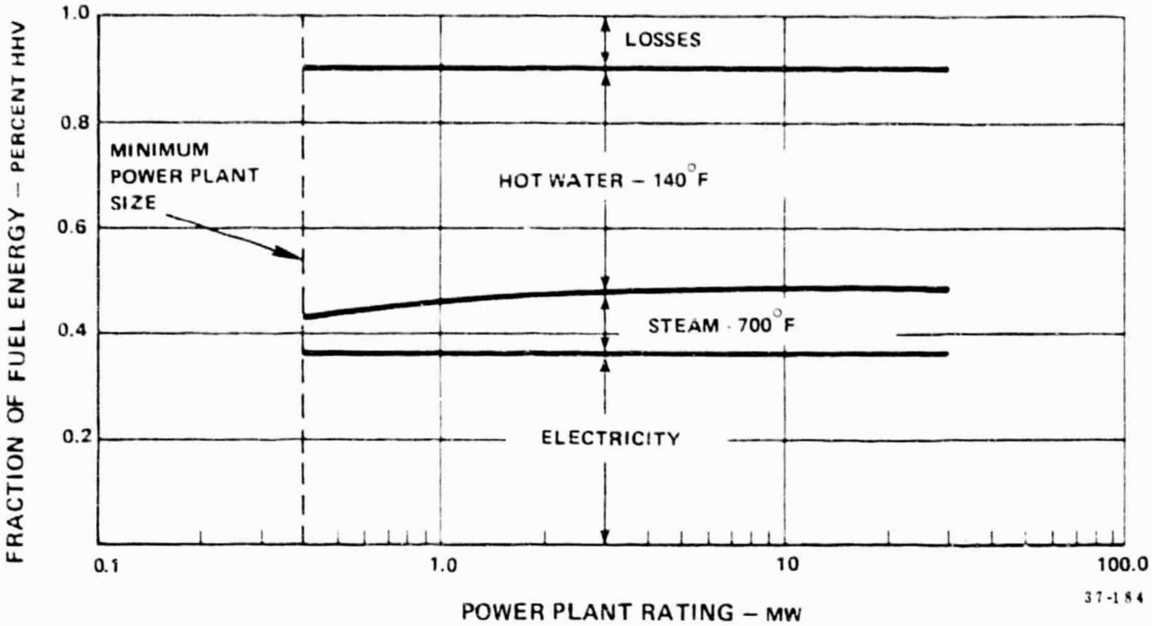


Figure III-121. High Temperature Fuel Cell Performance with Size - Coal-Derived Distillate Fuel - Conversion System 30 - Design Option 2

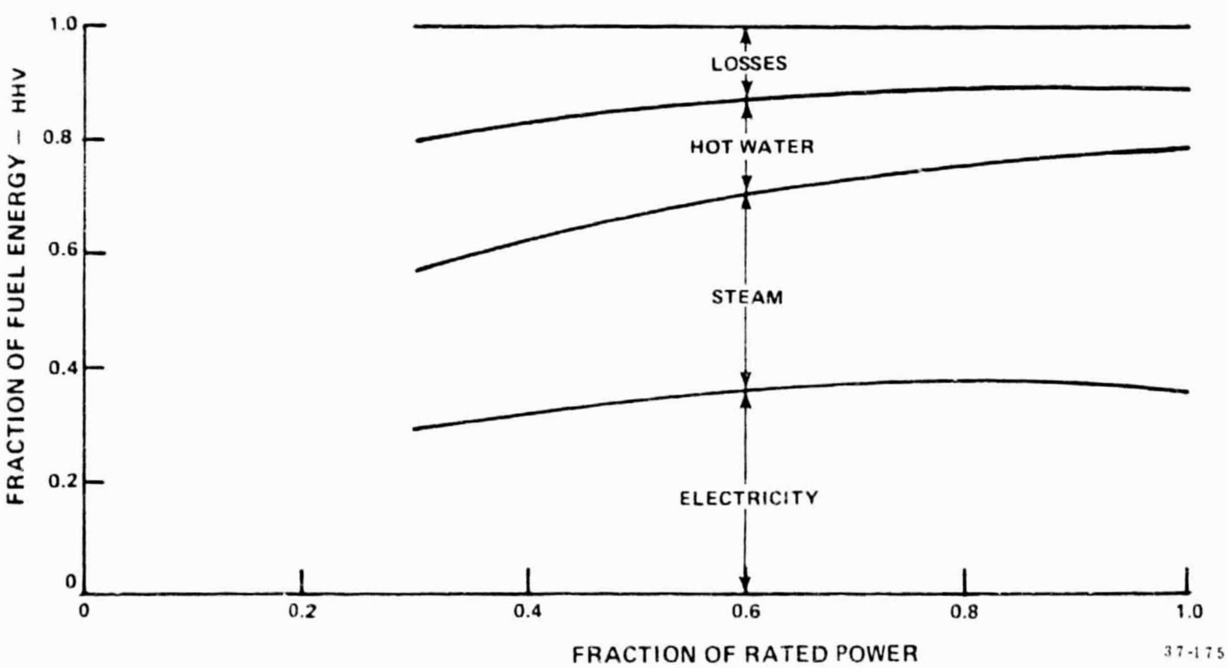


Figure III-122. High Temperature Fuel Cell Off-Design Performance - Coal-Derived Distillate Fuel - Conversion System 30 - Design Option 3

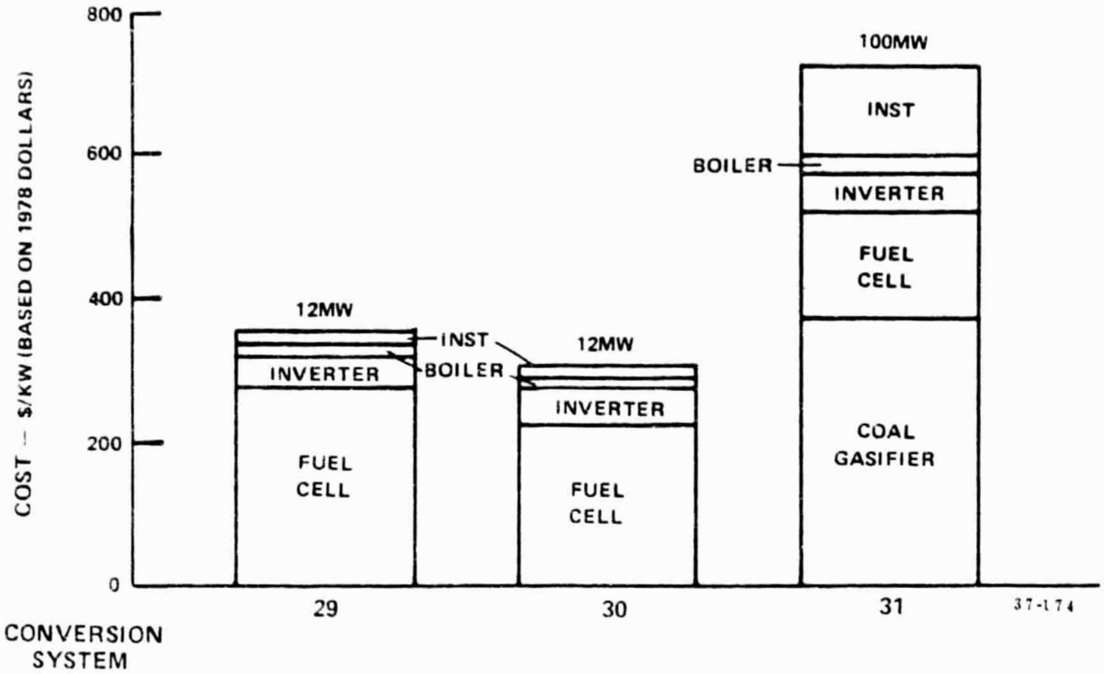


Figure III-123. High Temperature Fuel Cell Estimated Costs

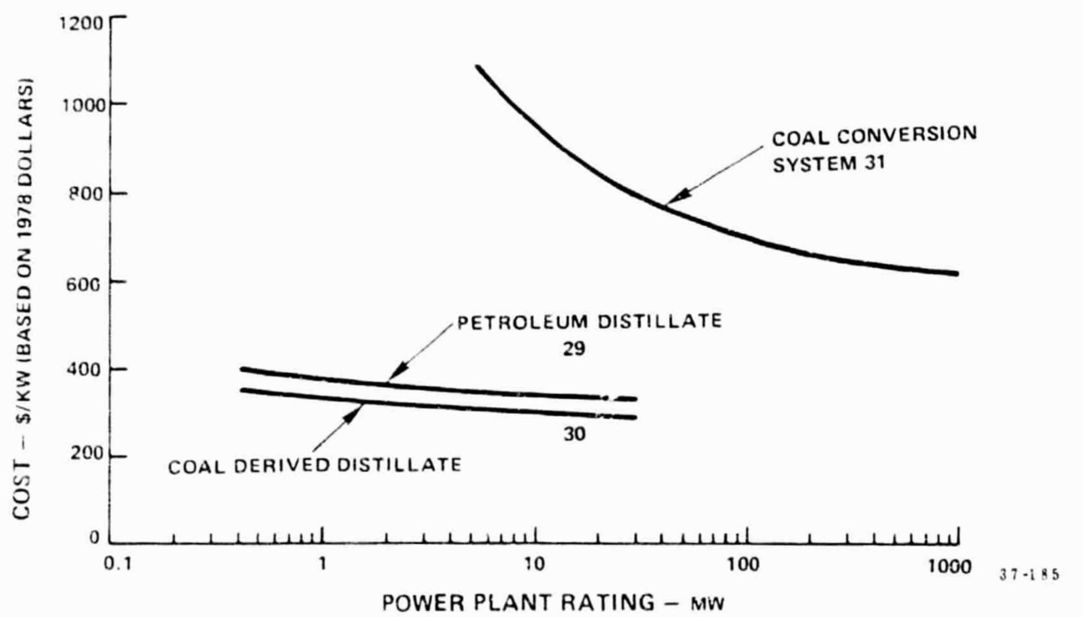


Figure III-124. High Temperature Fuel Cell Estimated Costs with Size

STIRLING ENGINES

INTRODUCTION

The Stirling engine is a versatile, regenerative closed-cycle piston engine that may be used in a cogeneration system to produce electrical power and several temperature levels of thermal energy from a single thermal input. For the Stirling engine system described herein, Figures III-125 and III-126, hot air from either a coal-fired atmospheric fluidized bed or a coal-derived, residual-oil-fired furnace is converted into electric power, steam at 500°F, and hot water at 140°F. The object of this study was to determine the total fuel utilization that could be achieved with a Stirling engine and the cost of the cogeneration system.

The Stirling engine is a closed-cycle, regenerative piston engine with cyclic compression and expansion of the working gas at different temperature levels. Power is produced in the conventional manner by compression of the working gas at low temperature, heating the gas, expanding it at a high temperature, and then cooling it to a low temperature again. Since the alternating heating and cooling of the shuttle gas would inherently waste large quantities of heat, a matrix-type regenerator is placed between the hot and cold volumes. Heat is then stored in the regenerator as the gas moves toward the cold space (compression volume) and then released as the working gas returns to the hot space (expansion volume). The net effect of these cyclic processes is the conversion of heat into mechanical power.

A single-cylinder, displacer-type Stirling engine is shown in Figure III-127. The idealized pressure-volume diagram (PV) and temperature-entropy diagram (TS) are given in Figure III-128. Together, these figures define the operation of a Stirling engine as described in Figure III-129, where:

a) Process A - B is an isothermal compression where heat is transferred from the working fluid at the minimum cycle temperature to the external

sink (heat rejection from the engine). This is described physically in Figure III-129 as the piston moves from its bottom dead center position (a) to its top dead center position (b).

b) Process B-C is at constant volume where heat is transferred to the working fluid from the regenerator matrix (heat released) as the displacer moves from its top dead center position (b) to its mid-stroke position. (c).

c) Process C-D is an isothermal expansion where heat is transferred to the working fluid at the maximum cycle temperature from the external source (heat added to the engine) as the displacer and piston move from the displacer mid-stroke position (c) to its bottom dead center position (d).

d) Process D-A is at constant volume where heat is transferred from the working fluid to the regenerative matrix (heat absorbed) as the displacer moves from its bottom dead center position (d) to its top dead center position (a), displacing gas from the expansion volume (hot space) to the compression volume (cold space).

If the heat that is transferred to the working fluid from the regenerator matrix is the same as that transferred from the fluid to the matrix, then only the external heat transfer processes remain, and the efficiency is consistent with the Carnot cycle efficiency. The advantage of the Stirling engine cycle is that the two isentropic processes of the Carnot cycle are replaced by two constant volume processes, thereby increasing the area under the PV diagram, reflecting in higher specific work output levels without resorting to very high pressures and high swept volumes.

CONVERSION SYSTEM DESCRIPTION

The two Stirling engine cogeneration systems studied are shown in Figures III-125 and III-126. The total system in Figure III-125 (energy conversion system number 32) consists of a coal-derived, residual-oil-fired furnace system designed by Bechtel National and described in Volume IV, the Stirling engine system designed by Mechanical Technology, Incorporated, and an interfacing hot air heat exchanger designed by Bechtel National. In Figure III-126, (energy conversion system number 33) the Stirling engine system interfaces directly with a hot air stream generated in the coal fired atmospheric fluid bed furnace. In each case, the Stirling engine system consists of the hot-end heat exchanger to transfer heat from the hot air stream to the working fluid of the engine, an intermediate heat exchanger to provide additional heat input into the working fluid at 1000°F, a hot water generator where heat is rejected from the engine, the engine block containing the pistons and regenerative heat exchangers, an intermediate boiler to produce 500°F steam and an electric generator. The use of an intermediate heat exchanger to add heat to the working fluid at an intermediate temperature, and the generation of work from this heat input is a Mechanical Technology Incorporated innovation that adds to the versatility of the Stirling engine for cogeneration. Its functioning and advantages will be defined.

The overall operating description of the engine is that heat supplied to the engine in the hot-end exchanger, and the intermediate heat exchanger, is converted into electric power, and the heat rejected from the engine is used to generate hot water. A description of the Stirling engine used in these systems and its operating characteristics follows.

Development of modern Stirling engines began in the late 1930's. Engines ranging from 1 to 400 horsepower have been built and studies have extended to 2000 horsepower. These designs serve as the background and basis for the larger machines considered is this study.

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The state-of-the-art is graphically portrayed in Figure III-130, where output power is plotted as a function of speed, mean cyclic pressure, and displaced volume. Engine designs below the curve are conservatively designed, and engines above the curve are advancements to the state-of-the-art. At present, those engines lying above the curve represent automobile engine designs where specific power output is a critical criterion and the engines must be designed to minimize size and weight. For stationary applications where greater emphasis is placed on reliability, maintainability, and durability, the curve more accurately reflects a true state-of-the-art.

A basic ground rule for this study was that the cogeneration systems designed would become commercially available after 1985; therefore, improvements in performance likely to occur before 1985 should be incorporated into the designs. For Stirling engines, future improvements in performance will come from two major advancements. These are: (1) raising the hot space temperature by the use of newly developed high temperature, high strength materials; and (2) by reducing the internal thermal and dynamic losses.

As indicated in Figure III-131, raising the temperature will lead to an increase in power and efficiency. For this program, a search was conducted for materials that could be used for Stirling engine heater heads at elevated temperatures from 1600 to 2000°F. Several materials were identified among the nickel-base super alloys and a partial list of such alloys, with the corresponding composition and characteristic rupture strength at 1600°F and 1800°F for periods up to 50,000 hours (5 years), was prepared. Data for these alloys generally do not exist to 50,000 hours, so an extrapolation from 1000 hours data (in some cases 20,000 hours data) was necessary. In Table III-50 these alloys are grouped according to those which must be fabricated by casting and those which exist in wrought form and can be fabricated by forging.

Table III-50 indicates that two alloys (IN-100 and IN-162) may have acceptable mechanical properties at 1600°F and a 10,000 psi stress level applied for 5 years, but that no alloys have adequate properties at 1800°F, even at 5000 psi applied for

the same period. It was judged from the study that to achieve temperatures above 1600°F would require the use of ceramics. To avoid the cost extrapolations that would have to be made if ceramic components were used, the study was limited to an upper temperature of 1600°F, which was considered readily achievable by the 1985 target year for the study.

The internal thermal and dynamic losses consist of all the loss mechanisms in the engine that produce a dissipation of thermal energy from the hot to cold end without the production of useful work. A discussion of these losses follows.

A) Regenerator Thermal Efficiency. Heat exchange in the regenerator is a periodic process between the gas flowing from the hot and cold spaces through the intermediate heat exchange with the regenerator matrix material. This loss represents a major loss mechanism for the system because the large temperature gradient across the regenerator means that a small regenerator inefficiency produces a large transport of heat out of the hot space.

B) Pressure Drop Loss. Pressure drop through the heat exchanger components and connecting passages irreversibly reduces the pressure amplitude in the engine that results from frictional flow losses and, therefore, an efficiency loss occurs.

C) Thermal Conduction Loss. Housing components extending between the hot and cold spaces provide a thermal path for the flow of heat out of the hot space to the cooler, where it is rejected without producing useful work.

D) Hysteresis Loss. Pressurization and depressurization of the working gas in the void spaces result in a heat exchange between the gas and the walls. The net effect is that energy leaves the working gas irreversibly and is dissipated through heat transfer to the environment.

The advancements in the performance of Stirling engines over the past 20 years are the result of the renewed interest in the machine that has led to the formulation of analytical tools that give insight into the physics of the engine. The expanded research funding in the field over the past few years, especially the emphasis being placed on high efficiency, low pollutant engines in the Department of Energy's "Stirling Engine Automotive Program", will surely further the advancement of Stirling engines. Furthermore, these advancements will be made by increasing the hot space temperature through the development of high strength, high temperature materials and a reduction of the losses described above.

The extrapolation of the increased power output as a function of the increase in heater head temperature is given in Figure III-131. Present machines typically run at about 1290°F; however, advanced designs now being considered will raise this temperature to 1560°F and, therefore, a performance improvement of 13 percent will occur. Based on this expectation and the expectation that additional improvements will also come from a reduction of parasitic losses, the dash curve in Figure III-130 is projected as the likely state-of-the art for Stirling engines by 1985. The dash curve was used in computing the efficiencies presented in this report.

The components considered as part of the Stirling engine cogeneration system are shown in Figure III-132. These components are:

A) Hot-End Heat Exchanger. A hot-end heat exchanger is used in the system to transfer heat from the furnace hot air stream to the working fluid. For the coal-derived, boiler fuel case, the inlet air to the heat exchanger was specified at 1800°F and a maximum average working gas temperature of 1600°F was used. This upper limit was imposed by material consideration as previously discussed. For the atmospheric fluidized bed furnace, the inlet air temperature was 1500°F, and an average working gas temperature of 1450°F was used to evaluate engine performance.

B) Steam Generator. The hot air exhaust from the hot-end heat exchanger is used in the steam generator to produce saturated steam at 500°F.

C) Intermediate Heat Exchanger. The intermediate heat exchanger is included to transfer residual heat in the hot air stream to the working gas to provide additional work output from the engine. By introducing this heat exchanger and the concept of a two-stage Stirling engine, it was felt that the versatility of the Stirling engine for a cogeneration application was increased, and advantage could be taken of the residual heat to increase the mechanical power output.

D) Hot Water Generator (Cooler). The hot water generator is used to reject heat from the Stirling engine. One of the energy requirements assumed for this study was hot water at 140°F; therefore, this temperature level was selected as the heat rejection temperature for the engine. For the performance analyses, the average temperature of the working fluid in the cooler was taken as 150°F.

E) Engine Block. In this system, the engine block consists of the piston crank mechanism and regenerative heat exchangers. The heat exchangers described above are external to the block.

The basic configuration for the Stirling engine is given in Figure III-133. The United Stirling AB engine was used to illustrate the adaptation of a Stirling engine to a cogenerative system. The difference in a large cogeneration engine would be in the hot air circuit, where the hot-end and intermediate finned-tube heat exchangers would be replaced with tube and shell heat exchangers because of the larger heat requirements and associated relative costs.

The engine design selected for this study is a two-stage engine where heat is added at the high temperature and also at an intermediate temperature. Work is produced through the addition of heat at each stage by staging the piston as shown in Figure III-133, and having an expansion volume at each temperature level. The conversion of heat to work in a two-stage Stirling engine is described in Figure III-134. A description of the cyclic process follows:

Position A. The displacer is in its uppermost position, and the power piston is slightly above its neutral position. At this point all of the working fluid is, ideally, in the compression volume (this neglects volume in the heat exchange components) at the mean cycle pressure, P_o, and the lowest cycle temperature T_c.

Position A to B. The displacer moves down, and the piston moves up through its top dead center position. This process, which is nearly a constant volume process, displaces the gas from the compression volume (V_c) to the two expansion volumes (V'_e and V''_e). Displacing the gas from V_c to V_e raises the temperature of the gas as it flows through the regenerators and heaters and, thereby, increases the pressure in the system to P_H.

Position B to C. The displacer and power piston move down together until the piston moves through its lowest position. This increases the system volume and displaces all of the gas into the expansion volumes, producing an expansion of the gas to P_o.

Position C to D. The expansion process continues, essentially at constant system volume, until the lowest cycle pressure is reached.

Position D to A. During this process the system volume decreases, displacing the gas back into the compression volume. During this process and process A to B, the heat of compression is rejected in the cooler as the gas is displaced from the compression to the expansion volume.

The configuration described above is a single-acting system in which the displacer and power piston are contained in the same cylinder. The configuration shown in the illustrative engine, Figure III-133, is a double-acting machine where the expansion and compression spaces are contained in adjacent spaces. This design has the advantage of compactness.

Figure III-135 describes a double-acting engine in which adjacent cylinders are connected together through a duct that connects the expansion space of one cylinder to the compression space of the adjacent cylinder. The connecting duct contains the heat transfer apparatus, specifically a heater, regenerator, and cooler. Each combination of expansion and compression space in adjacent cylinders, along with the associated heat exchangers contained in the connecting duct, forms a basic two-volume Stirling engine system.

The principle of operation of the double-acting machine is the same as the single-cylinder machine in that if heat is supplied at high temperature to the working fluid in the expansion space and abstracted at ambient temperatures from the working fluid in the compression space, then the net force acting over the whole cycle on the upper face of the piston in the expansion volume will be greater than the net force on the lower face of the piston in the compression volume. By connecting several cylinders together, the sum of the force acting on each piston will produce a mechanical work output from the engine.

The engine is defined by the boundaries specified in Figure III-136. For the thermodynamic analysis of the engine, the energies defined at the boundary of the system are the heat input at the high temperature (T_h), the intermediate temperature (T_e), the heat rejected to the hot water stream at 140°F, and the mechanical work supplied to the generator. For this system, the engine efficiency is defined as:

$$\eta_E = \frac{W_{SHAFT}}{Q_{HEAT}} = \frac{Q_{HEAT} - Q_{REJ}}{Q_{HEAT}} \qquad (2)$$

where: W_{SHAFT} = The shaft output work from the engine.

Q_{HEAT} = Heat added to the engine at the high and intermediate temperature levels.

Q_{REJ} = Heat rejected from the engine to the 60°C (140°F) sink.

In Equation (2), the engine efficiency is defined as the product of the thermal efficiency (η_{to}) and the mechanical efficiency (η_m), where η_m defines the loss in the engine due to crank and other frictional transmission losses.

In developing the Stirling engine for this application, the two-stage engine, as described in Figure III-133, was selected as a reference design because of its compatibility with a cogeneration application and its adaptability for converting the residual heat in the hot air stream to additional work. Two additional thermodynamic advantages of this system are:

a) Splitting the regenerators and establishing an intermediate thermal station with the residual heat allows the regenerator to be sized independently for the hot and cold regions, rather than having a single unit spanning the full temperature range. This has the advantage of optimizing the regenerator for pressure drop and heat transfer characteristics over each range and, therefore, overall regenerator efficiency will be improved.

b) All heat losses (regenerator, conduction, and hysteresis) occurring in the hot region become a heat input to the intermediate stage and, therefore, some work can be developed from these losses without their total rejection from the system.

The ideal temperature entropy diagram for the two-stage machine is shown in Figure III-137. The processes for this system are:

Process 1-2 Isothermal compression of the total working fluid (1 + X).

Process 2-3 Constant volume heating of the gas in the first stage regenerator.

Process 3-6 Isothermal expansion of a fraction (X) of the gas at the intermediate temperature level.

Process 3-4 Constant volume heating of the gas in the first-stage regenerator

Process 4-5 Isothermal expansion of the gas at T_h.

Process 5-6 and 6-1 Constant volume cooling of the gas in the two regenerators.

Predicted temperature distributions in the engine are shown in Figure III-138. These curves give the extreme cyclic temperature changes for the working fluid during one complete cycle. Following the distributions from left to right, gas leaving the compression volume flows through the cooler where the heat of compression is removed from the gas. The gas then travels through the first-stage regenerator where its temperature is raised by the periodic heat transfer with the counterflow gas stream. Next, heat is added to the gas in the intermediate heat exchanger. The gas stream then splits, a small fraction flowing into the intermediate heat exchanger and the remainder flowing up through the hot-end regenerator into the hot-end heat exchanger. In the hot-end heat exchanger, heat is added to the working fluid raising its temperature before it enters the hot-end expansion volume. After the expansion process, the flow reverses its path and returns to the cold-end space, completing the cycle.

The fuel utilization efficiency for a cogeneration system is defined as:

$$\text{Fuel Utilization} = \frac{\text{Energy Utilized}}{\text{Heating value of Fuel}}$$

For the Stirling system components, the total fuel utilization was defined from the boundary relationships shown in Figure III-139; a description of the energies follows:

Q_{IN} = The heat input to the system from the hot air stream from the furnace.

Q_{STEAM} = The heat transfer from the hot air stream to produce steam at 500°F

Q_{RETURN} = Heat returned to the furnace by the hot air stream, referenced from $T_{ambient}$

$Q_{HOT\ WATER}$ = Heat rejected by the engine to water at 140°F

Q_{LOSS} = All thermal energy lost to the ambient

W_{ELECT} = Electric power produced by the generator

The system energy balance is:

$$Q_{IN} - Q_{RETURN} = Q_{STEAM} + Q_{HOT\ WATER} + Q_{LOSS} + W_{ELECT}$$

By substituting the following equations:

$$Q_{CYCLE} = Q_{IN} - Q_{RETURN}$$

$$\eta_g = W_{ELECT}/W_{SHAFT}$$

$$\eta_E = \frac{W_{SHAFT}}{W_{SHAFT} + Q_{REJ}}$$

$$Q_{REJ} = Q_{HOT\ WATER} + Q_{LOSS}$$

We obtain:

$$\frac{Q_{STEAM}}{Q_{CYCLE}} + \frac{W_{ELECT}}{Q_{CYCLE}}\left[\frac{1}{\eta_E} - \frac{1}{\eta_g} + 1\right] = 1$$

$$\frac{Q_{STEAM}}{Q_{CYCLE}} + \frac{Q_{HOT\ WATER}}{Q_{CYCLE}} + \frac{Q_{LOSS}}{Q_{CYCLE}} + \frac{W_{ELECT}}{Q_{CYCLE}} = 1$$

Thus, by specifying the required electrical output and calculating the losses, the fraction of steam and hot water produced is governed by the engine and generator efficiency.

c-3

PERFORMANCE

The object of this study was to determine the Stirling engine's potential for cogeneration applications in sizes ranging from 500 to 30,000 hp. To cover this broad range, a 6-cylinder, double-acting engine module was established as a reference design, and the power output from the system varied in two ways. First, for the range of output below 5000 hp, the power output was varied by changing the cubic inch displacement and engine speed. Second, for power levels above 5000 hp, the operating characteristics were held constant and a parallel combination of modules was used to increase the output. In performing this study, no operating or thermodynamic limits were identified that would prevent the design of a larger engine module than the reference design; however, from a cost standpoint, it was considered more economical to have a single standardized module of moderate size and use multiple systems to achieve larger power outputs.

The physical dimensions and operating characteristics for the design options considered are given in Table III-51 for a hot-end temperature of 1450°F, and in Table III-52 for a 1600°F hot-end temperature. In both tables, the reference design was Option 2, which had a 12-inch diameter bore, 6-inch stroke, and ran at 1800 rpm. The size and speed were chosen because of their similarity to present-day diesel engines that are in commerical service. The mean cyclic pressure of 1000 psi was chosen because it represented a good trade-off between output power and structural requirements. It was also concluded that specific power (horsepower per pound) would not be critical in a large stationary application and, therefore, a lower pressure could be used. In addition, helium could be used as the working gas instead of hydrogen, as used in the automotive engines.

The analytic procedure followed to compute the performance was based on the engine efficiency, Equation (2), where the output shaft power was taken from the projected 1985 power curve given in Figure III-13. Q_{IN} was computed using the classical Schmidt analysis where the temperature ratio was derived for a two-stage machine as:

$$\tau = \frac{(1 + X)T_c}{T_h + X\,T_i} \tag{3}$$

where: T_c = Cold Space Temperature

T_h = Hot Space Temperature

T_i = Intermediate Temperature

X = Split in the flows between the hot and intermediate expansion volumes

and the total expansion volume is the sum of the hot-end and intermediate temperature expansion volumes. Also for a two-stage engine, it can be seen from Equation (3) that the equivalent for the Carnot efficiency in a single-stage machine is:

$$\eta_T = \frac{(1 + X)T_c}{T_h + X\,T_i} \tag{4}$$

The above analysis was used to define the design point operating characteristics of the machine. The off-design point performance was computed from performance maps measured from engines at United Stirling. A normalized map for this engine is shown in Figure III-140.

The results of this study include both the performance and cost of Stirling engines. The performance is presented as a system performance where the total fuel utilization is plotted over the size range 373 kW (500 hp) to 10 MW (13,410 hp). The percentage of the total fuel utilization is defined as:

$$\frac{Q_{STEAM}}{HHV} + \frac{Q_{HOT\ WATER}}{HHV} + \frac{\Sigma W_{ELECT}}{HHV} + \frac{Q_{LOSS}}{HHV} = 1 \tag{5}$$

where HHV = higher heating value of the fuel

The performance results are given in Figures III-141, III-142, III-143, and III-144 for the two temperature levels, 1450°F and 1600°F, and for two electric power outputs. From Figure III-141, it is seen that at 1600°F the total fuel utilization at 10 MW is 80 percent. For this case, the individual values are:

$$\frac{\Sigma W_{ELECT}}{HHV} = 40 \text{ percent}$$

$$\frac{Q_{HOT\ WATER}}{HHV} = 24 \text{ percent}$$

$$\frac{Q_{STEAM}}{HHV} = 16 \text{ percent}$$

The off-design performance for these cases is given in Figures III-145, III-146, III-147, and III-148. Off-design point performance was computed by using United Stirling AB's, engine maps to determine the efficiency and power change as a function of rpm and mean operating pressure. At reduced power levels, it was found that efficiency of the engine was significantly reduced by lowering the pressure.

ESTIMATED COSTS

Cost for the Stirling engine was derived by extrapolating from known costs for similar diesel engines and tube and shell heat exchangers. The cost structure breakdown used was as follows:

A) Engine Block. The engine block was considered to consist of the housing, pistons, crank mechanisms, and regenerative heat exchangers. The cost of the housing, pistons, and crank mechanisms was found using the diesel engine data given in Figure III-149 which is consistent with the data in Table III-17. To this cost, an additional cost for the regenerative heat exchangers was added. Regenerator costs were computed from known material and handling costs for smaller units and scaled to these larger, multiproduced units.

B) Heat Exchangers. The heat exchangers included in this category were the hot-end heat exchanger (heater), the steam generator, the intermediate heat exchanger, and the cold-end heat exchanger (cooler). The costs for tube and shell heat exchangers for this application are given in Figure III-150. The costs given in this figure are a function of the heat transfer area, which can be computed from the performance data given in Tables III-48 and III-49, by assuming a temperature difference between the two exchange streams.

C) Electric Generator. The following cost data were used for the generators:

1,000 kW @ 1,800 rpm - $35.00 per kW
3,500 kW @ 1,800 rpm - $25.00 per kW

D) Controls. The control package considered for this program was only a basic diagnostic package to monitor essential system operating parameters and a pressure-modulating package to vary output power. It was assumed that speed would be controlled by synchronizing the generator output to line power. Based on these assumptions, it was estimated that the control package would be approximately 8 percent of the system cost.

Breakdowns of the Stirling engine costs are given in Tables III-50 and III-51 for the reference designs; the cost of the system as a function of size is given in Figure III-151. The higher cost for the high-temperature case reflects the higher cost for materials in the engine and hot-end heat exchanger. The costs of the heat sources and engine installation are presented in Volume IV.

EMISSIONS

The emissions from the heat sources and the waste materials discharged from coal-fired furnaces are defined in Volume IV.

PHYSICAL CHARACTERISTICS

The Stirling engine-generator is estimated to weigh 50 pounds per kilowatt and occupy about 0.06 square feet per kilowatt.

COGENERATION APPLICABILITY

The Stirling engine-generator is considered a promising candidate for cogeneration. It offers fuel flexibility with a separate heat source. The Stirling engine can operate at high electrical efficiency over a wide range of output levels. Response to changes in demand are governed by the response of the heat source. The Stirling engine-generator can be grid connected or operate independently. With the modular concept, additions to meet increasing process/plant requirements are straightforward. The basic power plant design can be modified to provide larger amounts of process steam at some penalty in electrical efficiency.

FUTURE DEVELOPMENTS

To achieve significant cogeneration use in the 1985-2000 period, the following technical developments are appropriate:

1. Development of high temperature materials with long term creep strength for the hot portion of the engine.

2. Development of configurations with low thermal losses from hot to cold end.

3. Development of high temperature heat exchangers and regenerators with high effectiveness.

4. Development of large, multi-cylinder Stirling engines with an improved combination of speed, mean cyclic pressure, and displaced volume.

5. Engineering development to reach required levels of capital cost and durability for commercially competitive equipment.

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3. A Collection of Stirling Engine Reports from General Motors Research - 1958 to 1970, General Motors Research Laboratories, Warren, Michigan, April 1978.

Table III-47. Potential High-Temperature Heater Head Materials

Designation	Composition	Characteristic Rupture Strength (ksi)* 1600°F 1000 Hrs.[1]	1600°F 50,000 Hrs.[2]	1800°F 1000 Hrs.	1800°F 50,000 Hrs.[2]
Cast Alloys					
IN-100	10% Cr, 15% Co, 3% Mo, 1% V 4.7% Ti and 5.5% Al, Bal. Ni	35.7	18.4 (12)*	15	6.6 (4.0)
IN-162	10% Cr, 4% (Mo, W, Cb) 1% Ti 6.5% Al, 2% Ta, Bal. Ni	34.6	19.7 (11.8)	16.5	7.7 (4.6)
Rene 85	9.3% Cr, 15% Co, 8.5% (Mo, W) 8% (Ti, Al), Bal. Ni	32.	13. (7.8)	10.	3.3 (2.0)
Wrought Alloys					
Udimet 700	15% Cr, 18.5% Co, 5% Mo 8% (Ti, Al), Bal. Ni	24.6	7.6[3] (4.6)	8.	2.4 (1.4)
Inconel 617	22% Cr, 12.5% Co, 9% Mo 1% Al, Bal. Ni	10	5.6[3] (3.4)	4.	2. (1.2)
Multimet (N155)	20% Cr, 18.5% Co, 20% Ni, 6% (Mo, W), 2% Mn, Bal. Fe	10	5.4 (3.2)	3.	1. (0.6)

1 Values interpolated between 1500°F and 1800°F data unless otherwise noted.
2 Values extrapolated from 1800°F data unless otherwise noted.
3 Based on 20,000 hour data.
* 1% creep stresses are shown in parentheses.

87-46

Table III-48. Stirling Engine Operating Characteristics

(T_h = 1450°F)

			REFERENCE DESIGN	
Physical Dimensions				
Maximum piston diameter	(ft)	0.5	1.0	1.0
Stroke	(ft)	0.25	0.5	0.5
Speed	(rpm)	1800	1800	1200
No. of cylinders per module		6	6	6
No. of modules		1	1	5
Total expansion volume	(ft^3)	0.05	0.39	0.39
Piston rod volume	(ft^3)	3E-4	0.03	0.03
Void volume	(ft^3)	0.150	0.79	0.79
Operating Parameters				
Mean cycle pressure	(psi)	1000	1000	1000
Pressure amplitude	(psi)	206	265	265
Pressure ratio (P_r)		1.5	1.7	1.7
Hot end temperature	(F)	1450	1450	1450
Intermediate temperature	(F)	500	500	500
Cooler temperature	(F)	150	150	150
Working fluid		Helium	Helium	Helium
Regenerator inefficiency	(%)	5	2	2
Output Characteristics per cylinder	(hp)	82	1000	600
Engine efficiency	(%)	38	51	48
Total Output Characteristics				
Transmission efficiency		95	90	90
Electric generator efficiency		95	97	97
Overall system efficiency		36	45	42
Total electric power output (KW)	(KW)	330	3902	11,707
$\Sigma W/Q_{cycle}$		.35	.35	.35
Q_{steam}/Q_{cycle}		.01	.26	.22
Q_{140}/Q_{cycle}		.54	.29	.33
Q_{loss}/Q_{cycle}		.10	.10	.10
Thermal Requirement				
Heat input (total for the cycle)	(MW)	0.9	11.1	33.4
Heat rejected to 60°C(140°F)water	(MW)	0.5	3.2	11.0
Heat input to the 260°C(500°F)steam	(MW)	0.01	2.9	7.3

37-61

Table III-49. Stirling Engine Operating Characteristics

(T_h = 1600°F)

			REFERENCE DESIGN	
Physical Dimensions				
Maximum piston diameter	(ft)	0.5	1.0	1.0
Stroke	(ft)	0.25	0.5	0.5
Speed	(rpm)	1800	1800	1200
No. of cylinders per module		6	6	6
No. of modules		1	1	5
Total expansion volume	(ft^3)	0.05	0.39	0.39
Piston rod volume	(ft^3)	3E-4	0.03	0.03
Void volume	(ft^3)	0.150	0.79	0.79
Operating Parameters				
Mean cycle pressure	(psi)	1000	1000	1000
Pressure amplitude	(psi)	206	265	265
Pressure ratio (P_r)		1.5	1.7	1.7
Hot end temperature	(F)	1600	1600	1600
Intermediate temperature	(F)	500	500	500
Cooler temperature	(F)	150	150	150
Working fluid		Helium	Helium	Helium
Regenerator inefficiency	(%)	5	2	2
Output Characteristics per Cylinder	(hp)	90	1100	660
Engine efficiency	(%)	45	56	53
Total Output Characteristics				
Transmission efficiency		95	90	90
Electric generator efficiency		95	97	97
Overall system efficiency		41	49	46
Total electric power output (KW)	(KW)	363	4292	13,599
$\Delta W/Q_{cycle}$		.35	.35	.35
Q_{steam}/Q_{cycle}		.16	.32	.28
Q_{140}/Q_{cycle}		.39	.23	.27
Q_{loss}/Q_{cycle}		.10	.10	.10
Thermal Requirement				
Heat input (total for the cycle)	(MW)	1.0	12.3	38.6
Heat rejected to 60°C(140°F)water	(MW)	0.4	2.8	10.4
Heat input to the 260°C(500°F)steam	(MW)	0.2	3.9	10.8

87-62

TABLE III-50. STIRLING ENGINE - GENERATOR EQUIPMENT COST ESTIMATES T_h = 1600°F

Electrical Output	4.3 MW
Shaft Speed	1800 RPM
Number of Cylinders	6
Maximum Piston Diameter	1 ft.
Stroke	0.5 ft.
Engine and Controls	\$180 per kW
Generator	25
Heat Exchangers	60
Total Cost	\$265 per kW

TABLE III-51. STIRLING ENGINE - GENERATOR EQUIPMENT COST ESTIMATES T_h = 1450°F

Electrical Output	3.9 MW
Shaft Speed	1800 RPM
Number of Cylinders	6
Maximum Piston Diameter	1 ft.
Stroke	0.5 ft.
Engine and Controls	\$145 per kW
Generator	25
Heat Exchangers	55
Total Cost	\$225 per kW

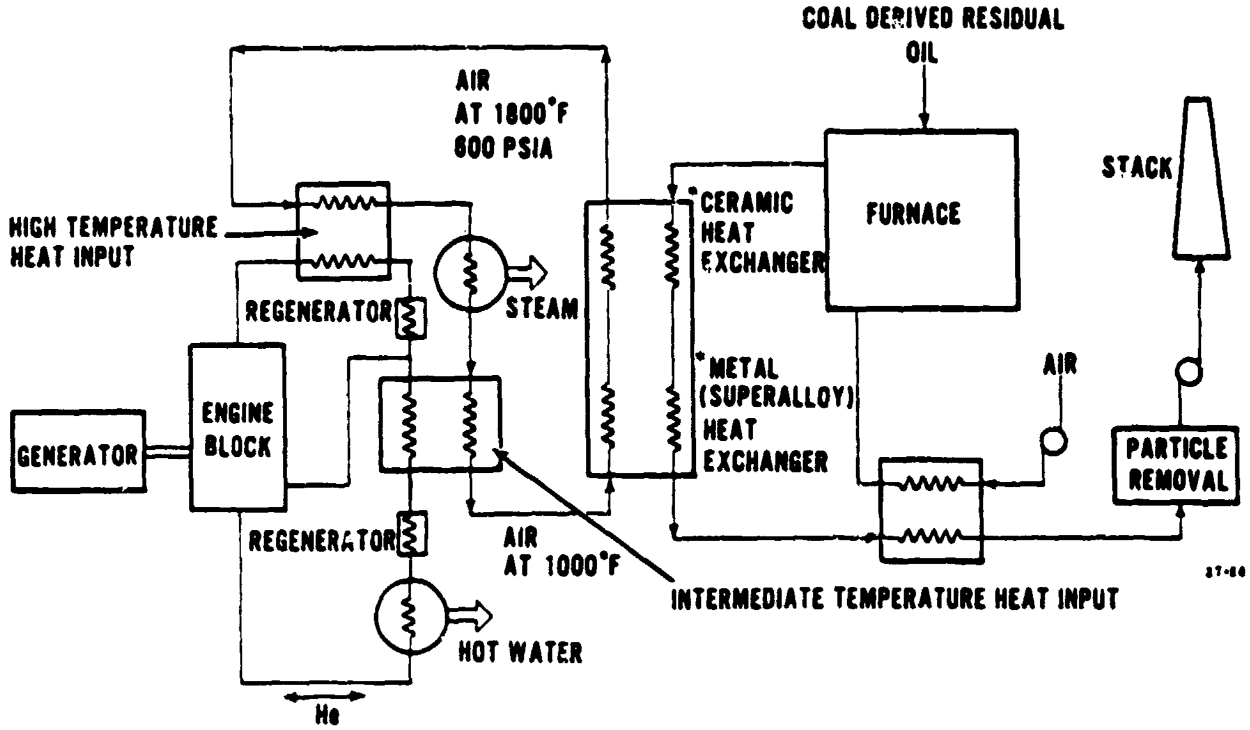


Figure III-125. Stirling Engine Hot Gas Furnace System

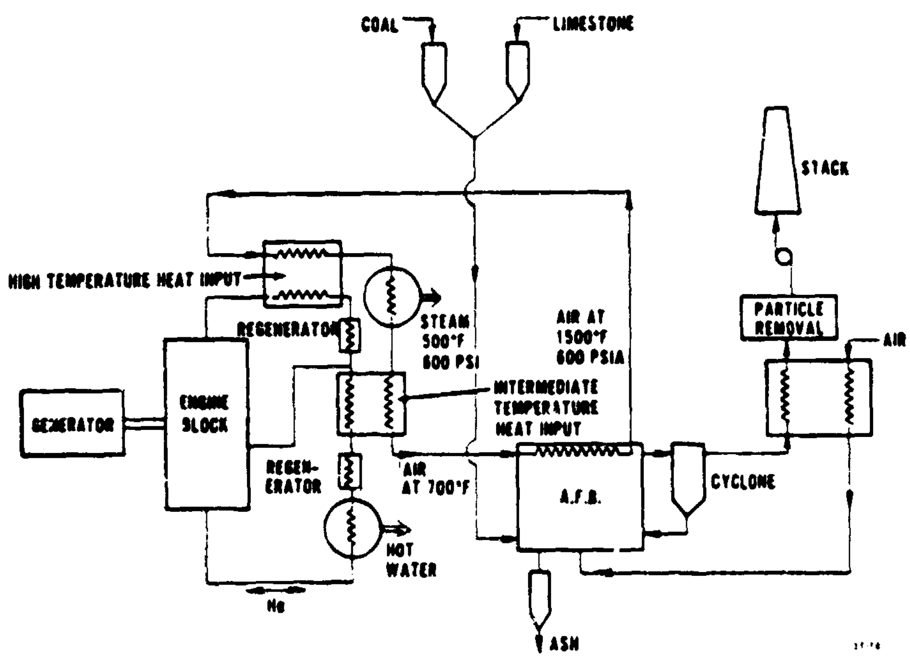


Figure III-126. Atmospheric Fluidized Bed Hot Gas Generator Stirling Engine System

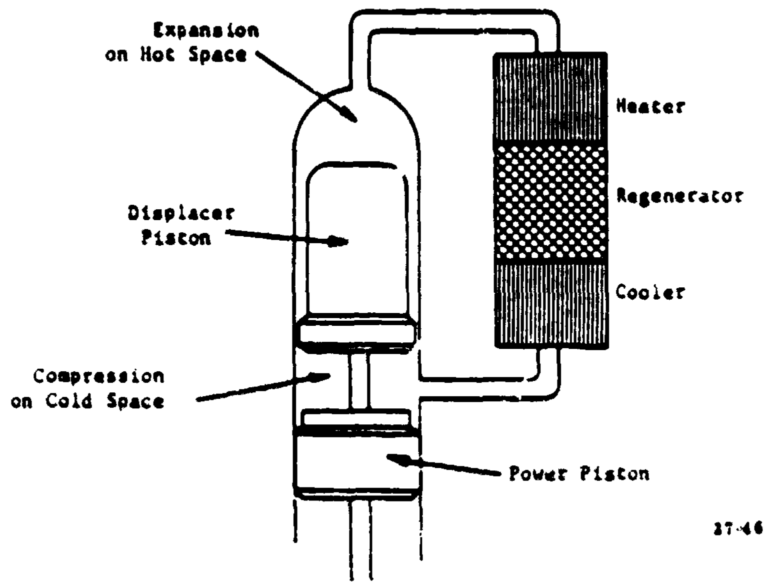


Figure III-127. Displacer or β Engine Configuration

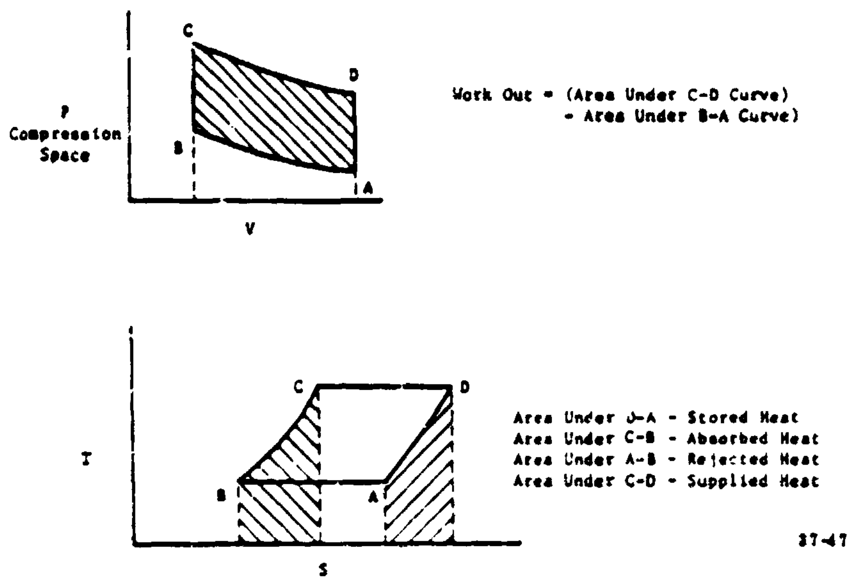
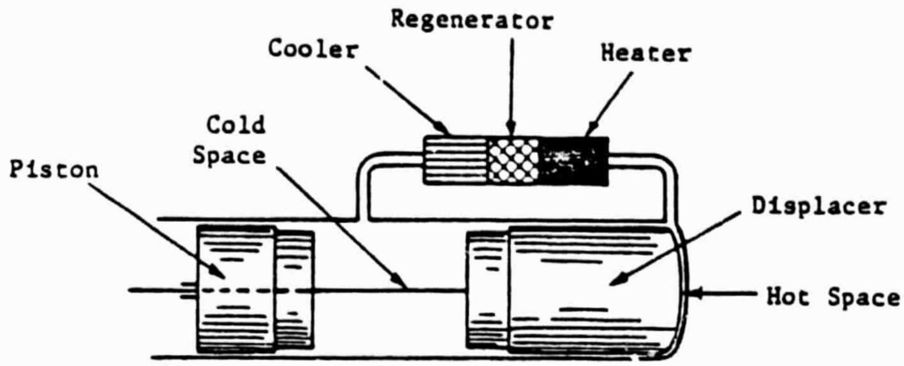
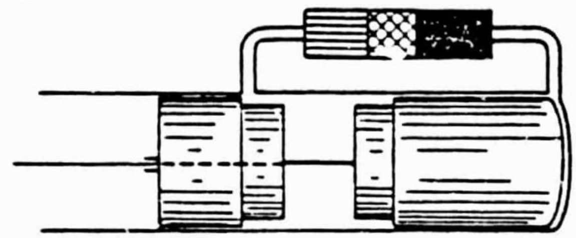


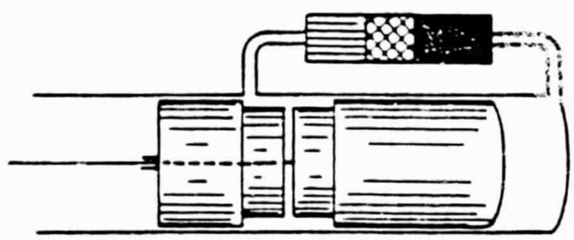
Figure III-128. Ideal Stirling Cycle P-V and T-S Diagram



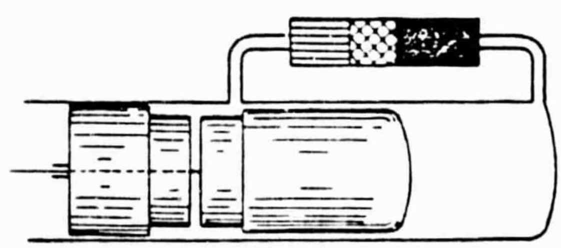
(a) Piston at bottom dead center. Displacer at top dead center. All gas in cold space.



(b) Displacer remaining at top dead center. Piston has compressed gas at lower temperature.



(c) Piston remaining at top dead center. Displacer has shifted gas through cooler regenerator and heater into hot space.



(d) Hot gas expanded. Displacer and piston have reached bottom dead center together. With piston stationary, displacer now forces gas through heater, regenerator and cooler into cold space, thus re-attaining situation (a) above.

37-70

Figure III-129. Typical Operation of a Displacer - Type Stirling Engine

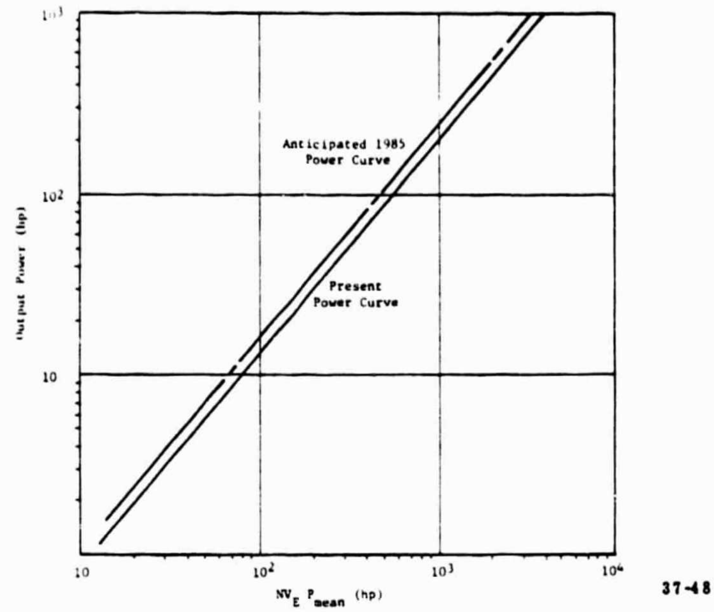


Figure III-130. Actual Power Output for Stirling Engines as a Function of the Speed, Expansion Volume and Mean Cycle Pressure

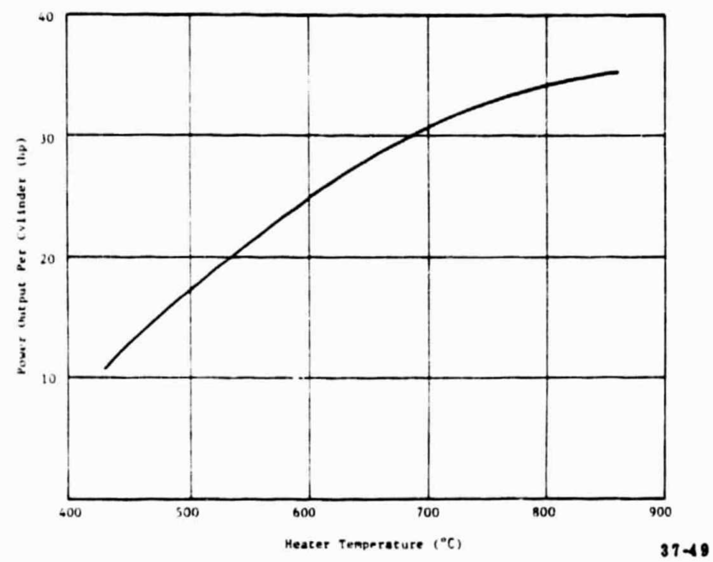


Figure III-131. Generalized Increase in Power Output From a Stirling Engine as a Function of the Heater Temperature

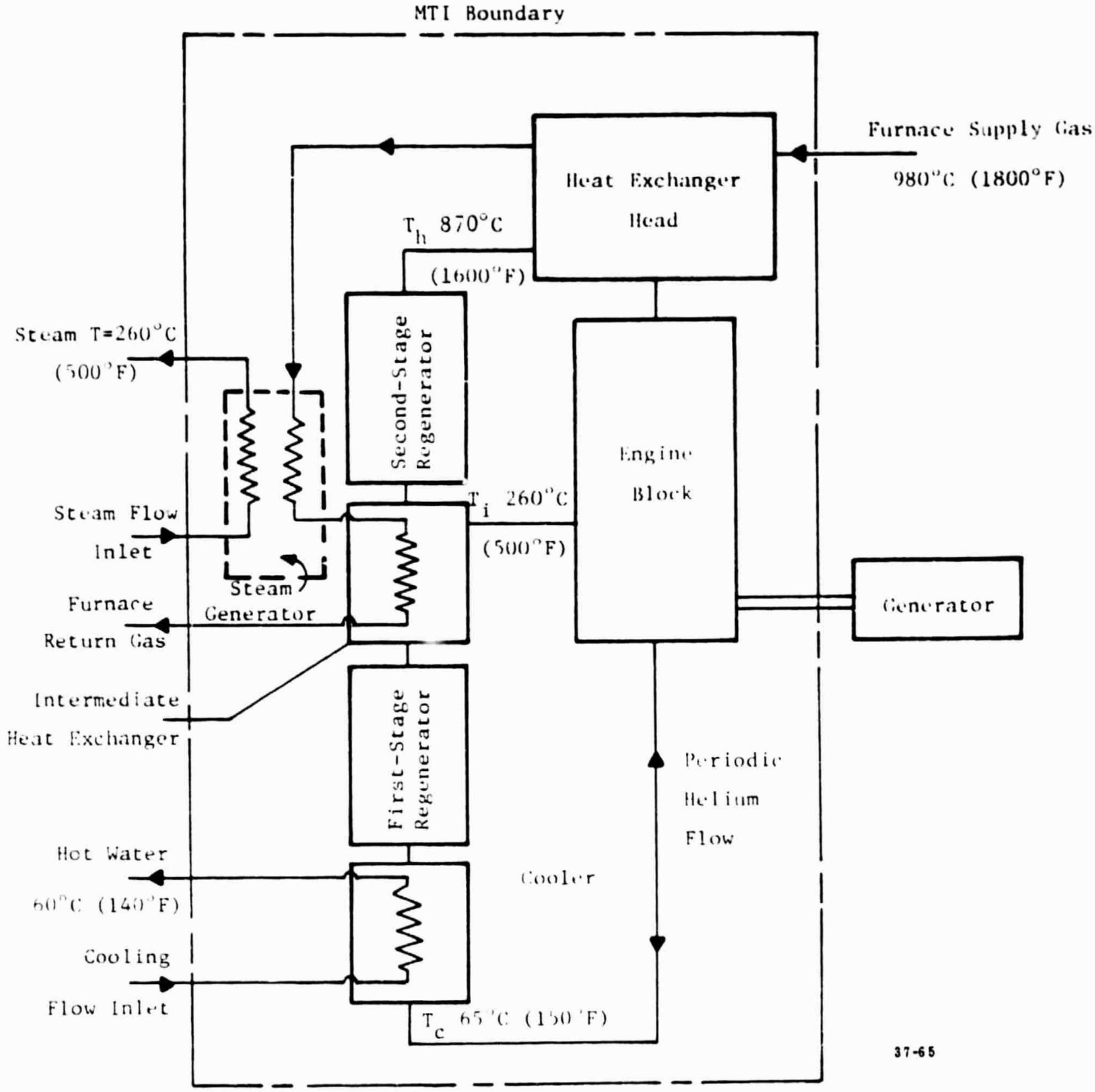


Figure III-132. Components for a Stirling Engine Cogeneration System

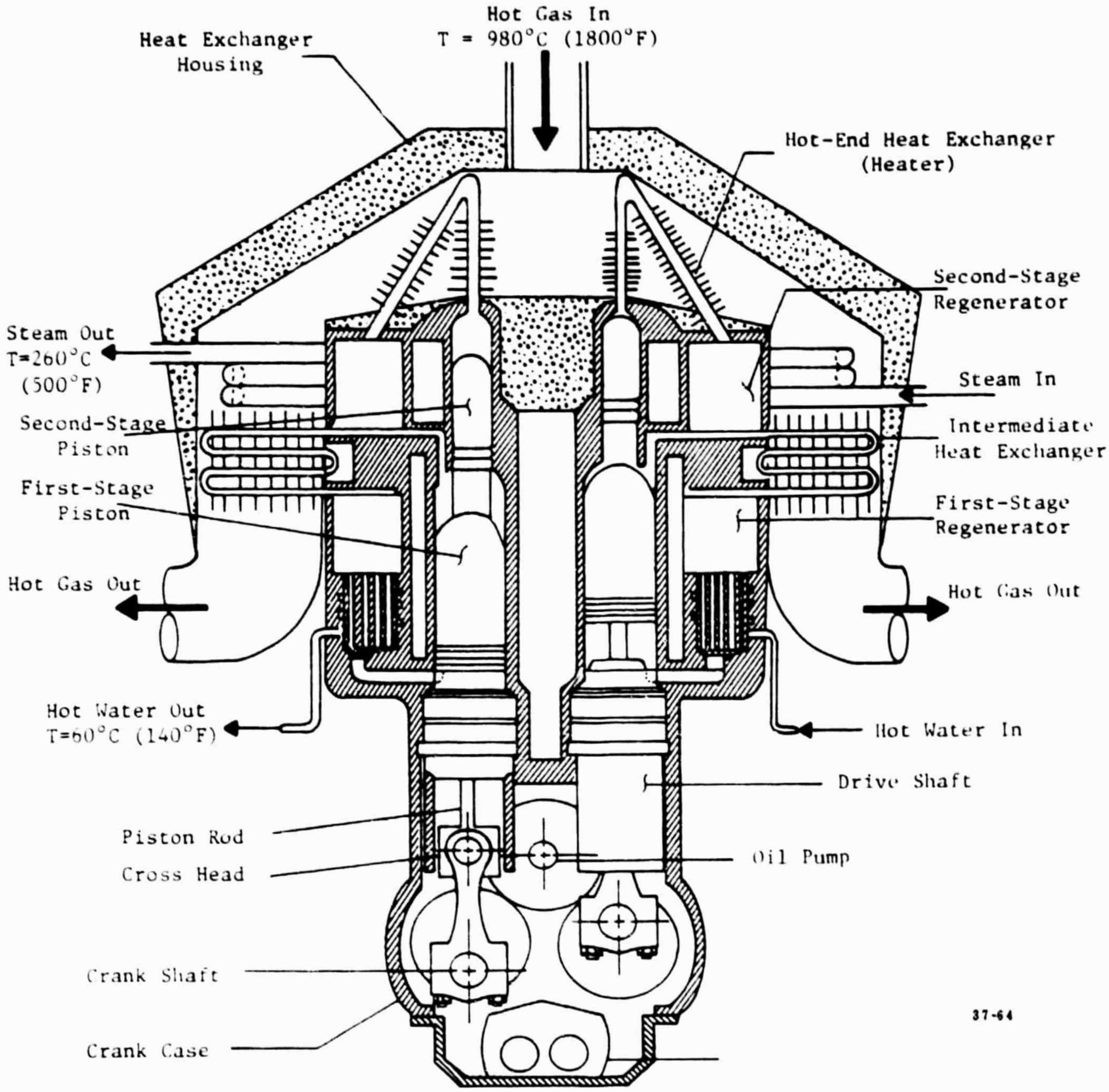
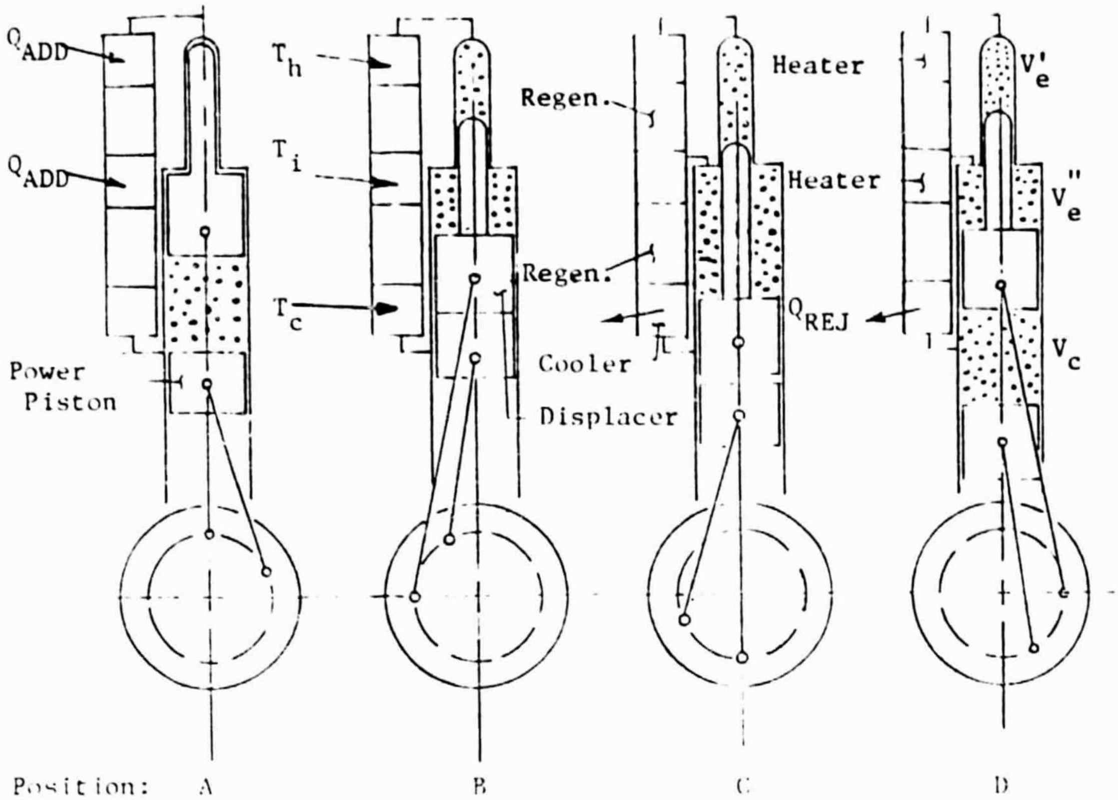


Figure III-133. Double-Crank, Double-Acting Stirling Cycle Engine Configuration



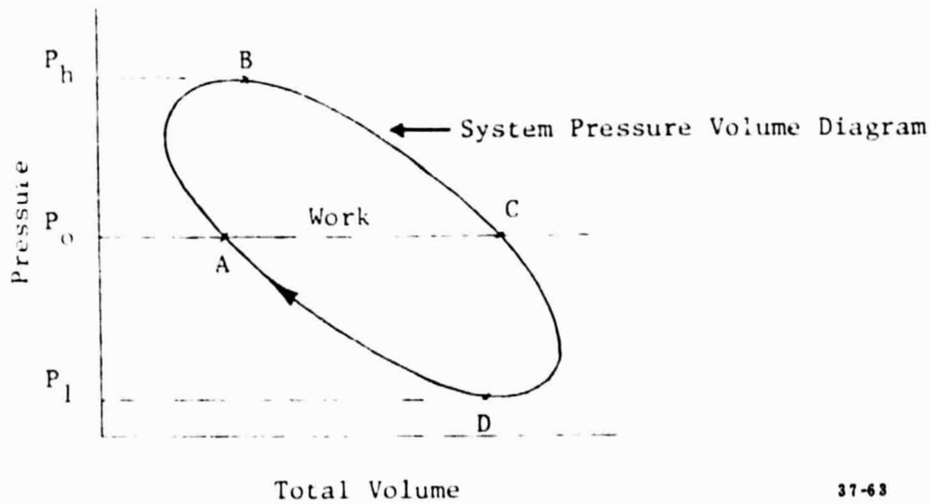


Figure III-134. System Dynamics for a Two-Stage Reciprocating Heat Engine

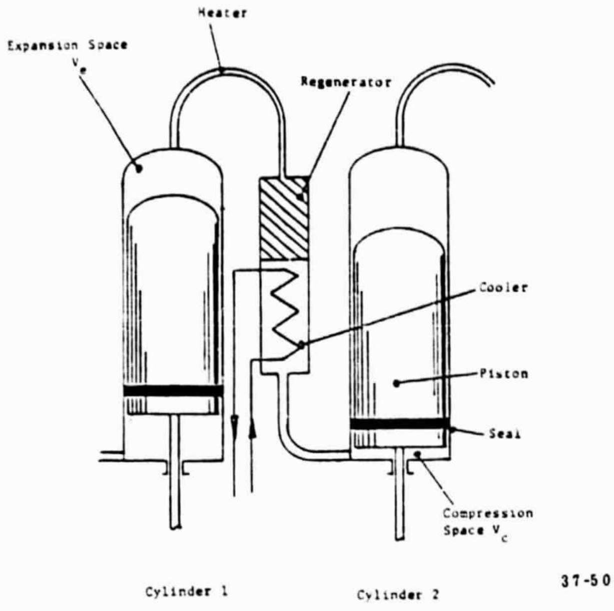


Figure III-135. Double-Acting Stirling Engine

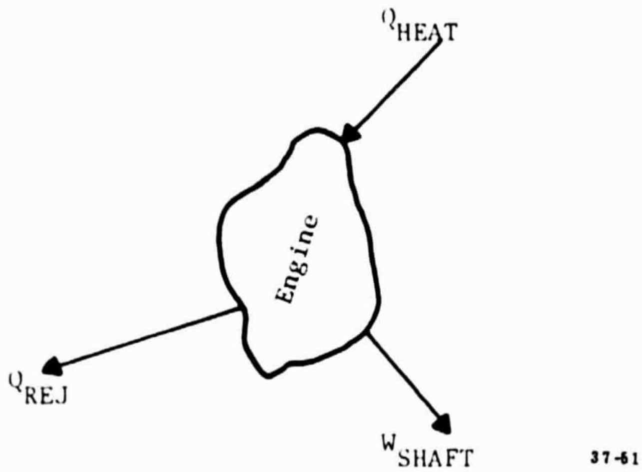


Figure III-136. Engine Boundary Definition

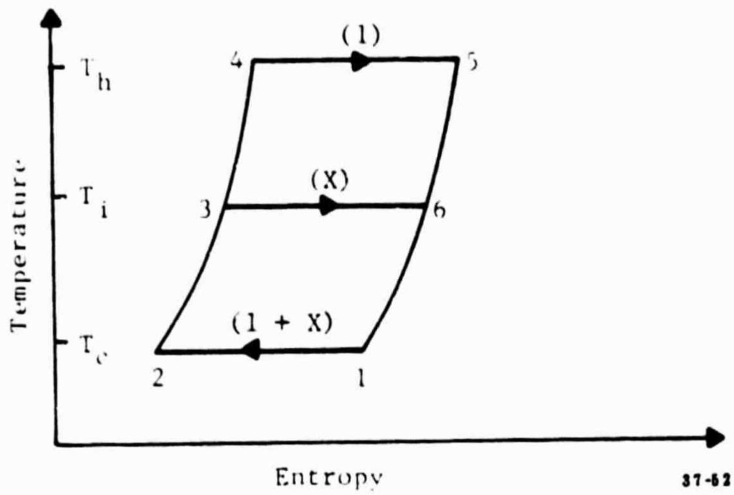


Figure III-137. Ideal Temperature Entropy Diagram for a Two-Stage Stirling Engine

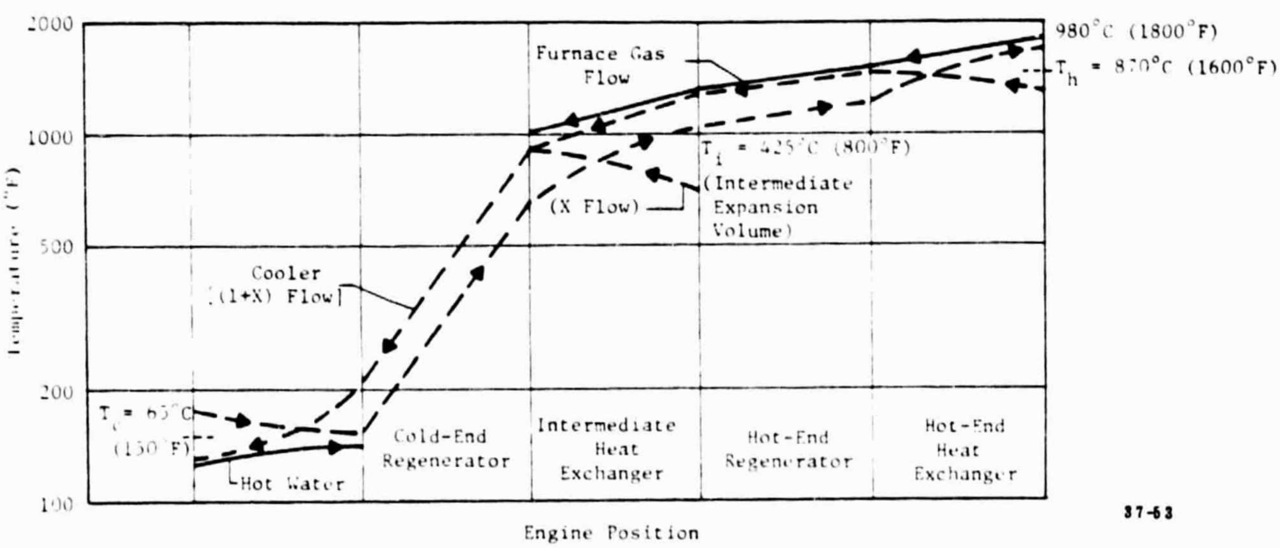


Figure III-138. Temperature Distributions Through a Two-Stage Stirling Engine

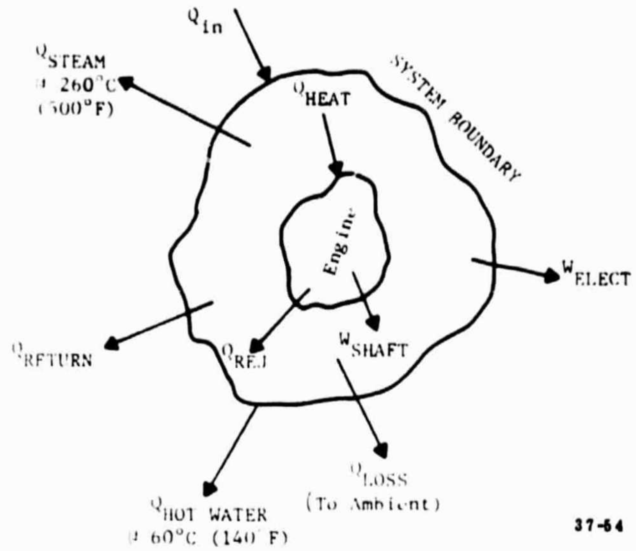


Figure III-139. System Boundary Definition

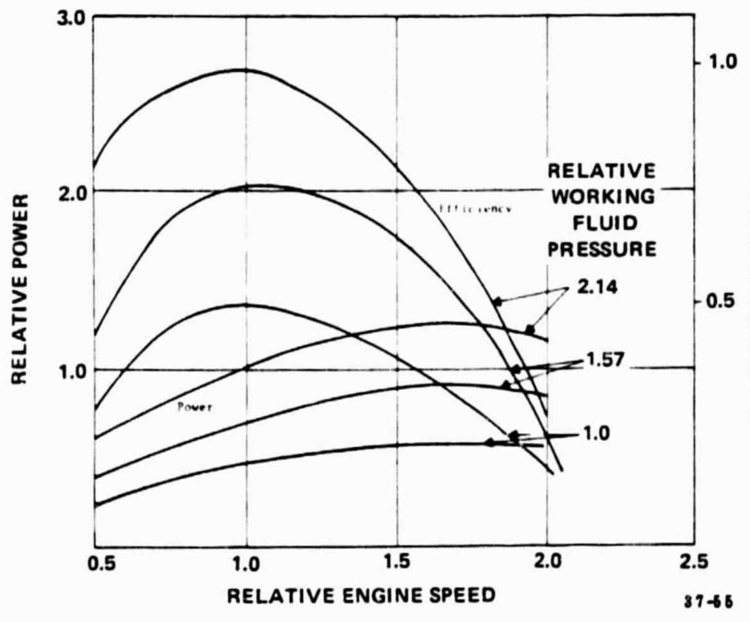


Figure III-140. Stirling Engine Power and Efficiency vs. Engine Speed

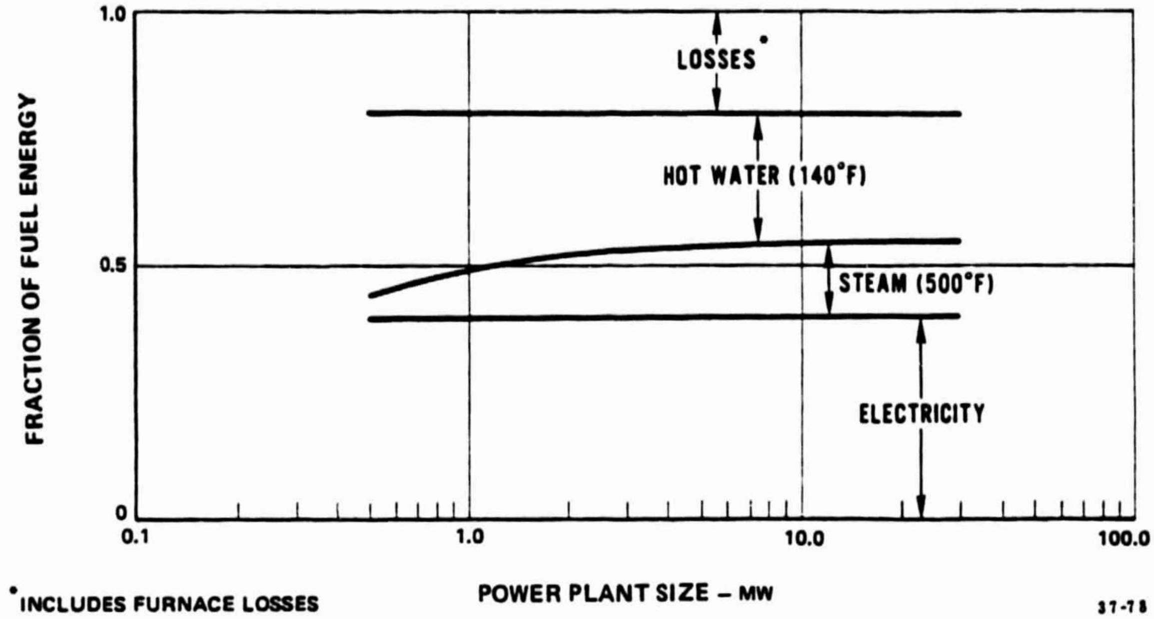


Figure III-141. Stirling Engine and Hot Gas Furnace Performance with Size T_h = 1600°F T_c = 150°F Design Option 1

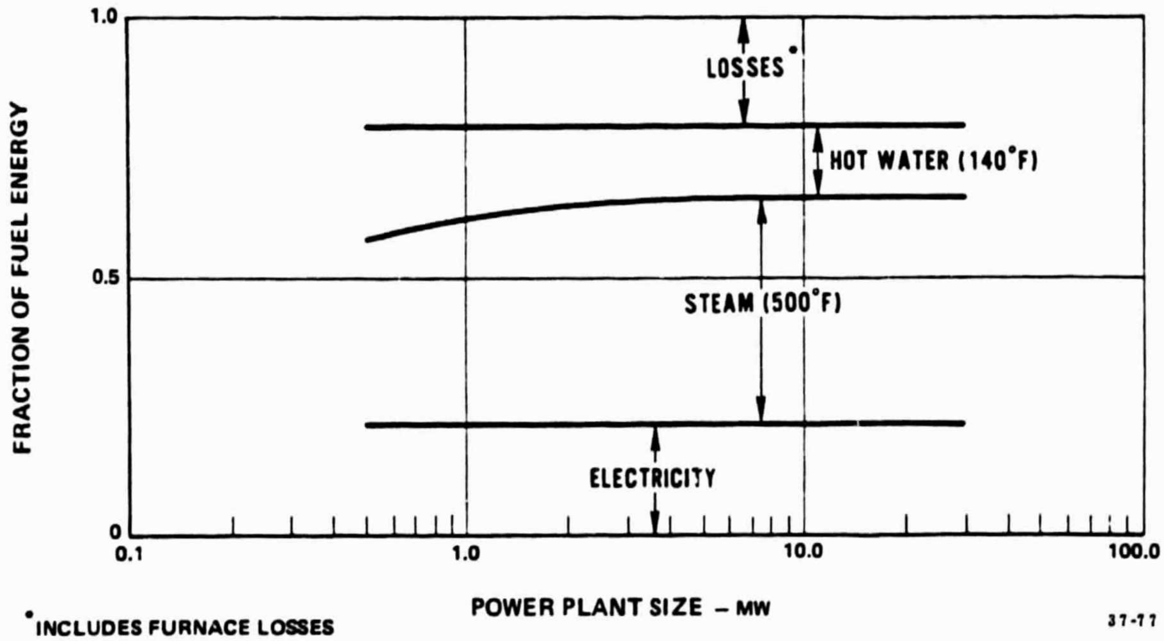


Figure III-142. Stirling Engine and Hot Gas Furnace Performance with Size T_h = 1600°F T_c = 150°F Design Option 2

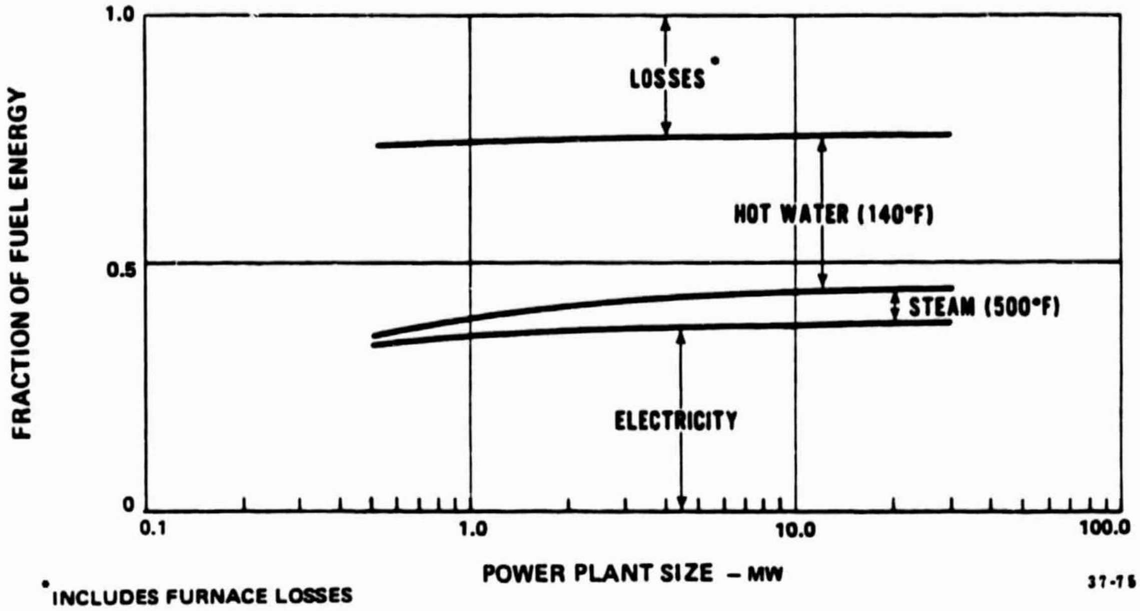


Figure III-143. Stirling Engine and Coal Atmospheric Fluidized Bed Heat Source T_h = 1450° F T_c = 150°F Design Option 1

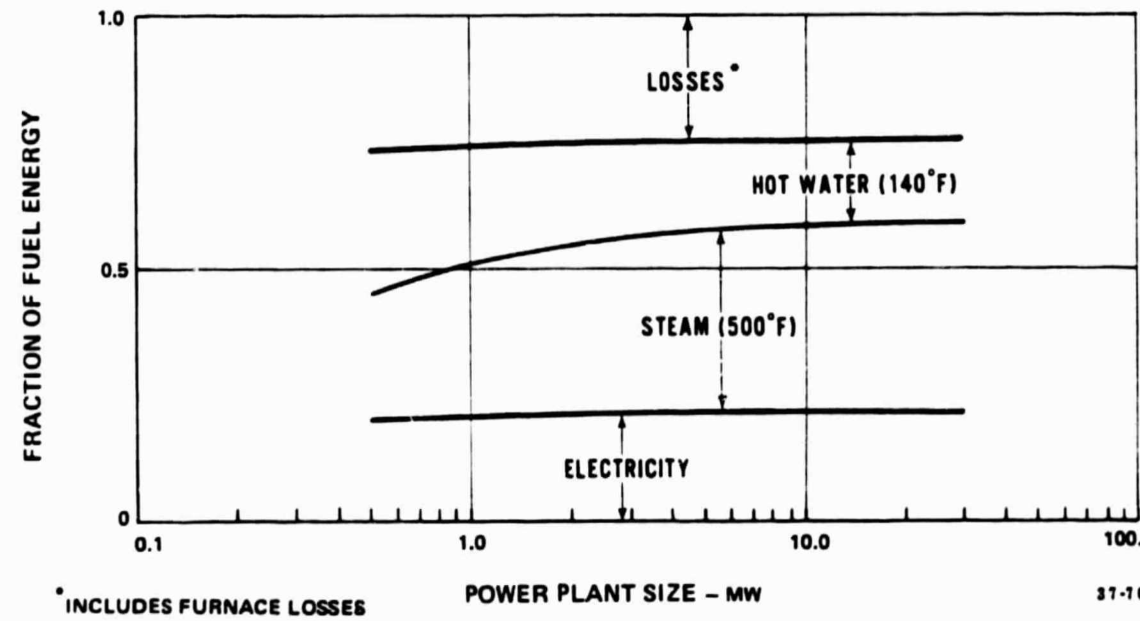


Figure III-144. Stirling Engine and Coal Atmospheric Fluidized Bed Heat Source T_h = 1450° F T_c = 150°F Design Option 2

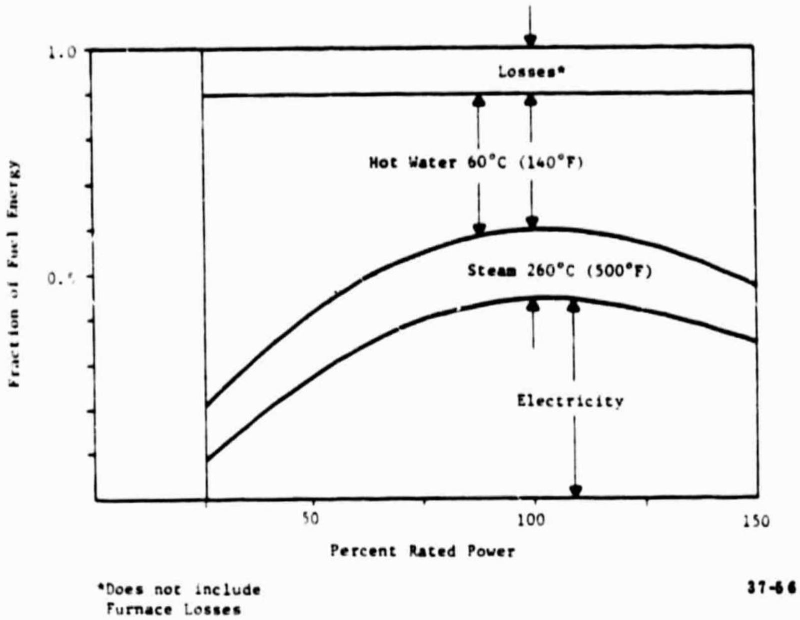


Figure III-145. Off-Design Performance of Stirling Engine and Hot Gas Furnace T_h = 1600°F Design Option 1

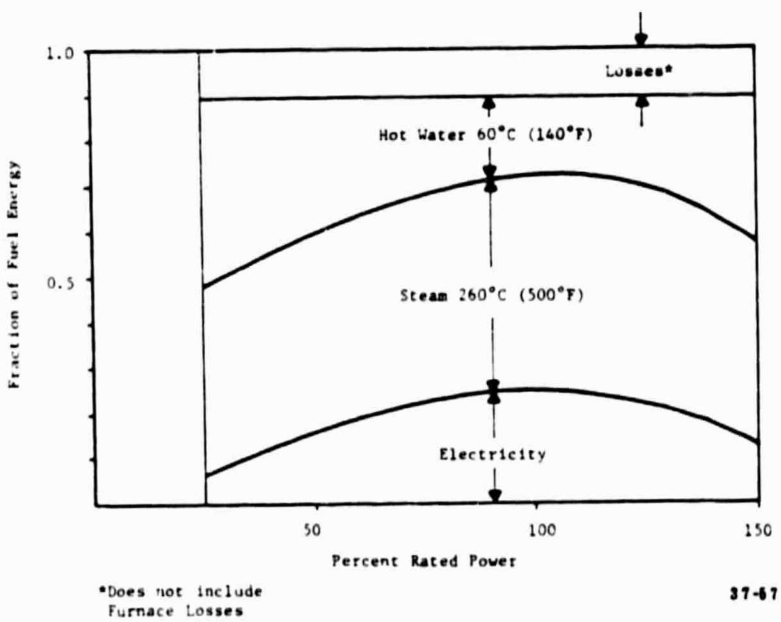


Figure III-146. Off-Design Performance of Stirling Engine and Hot Gas Furnace T_h = 1600°F Design Option 2

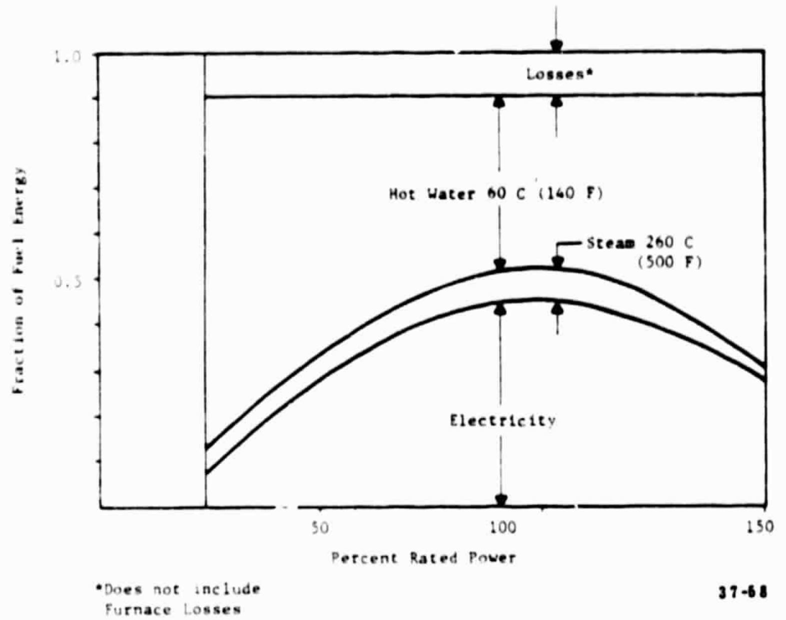


Figure III-147. Off-Design Performance Stirling Engine with Coal Atmospheric Fluidized Bed T_h = 1450°F Design Option 1

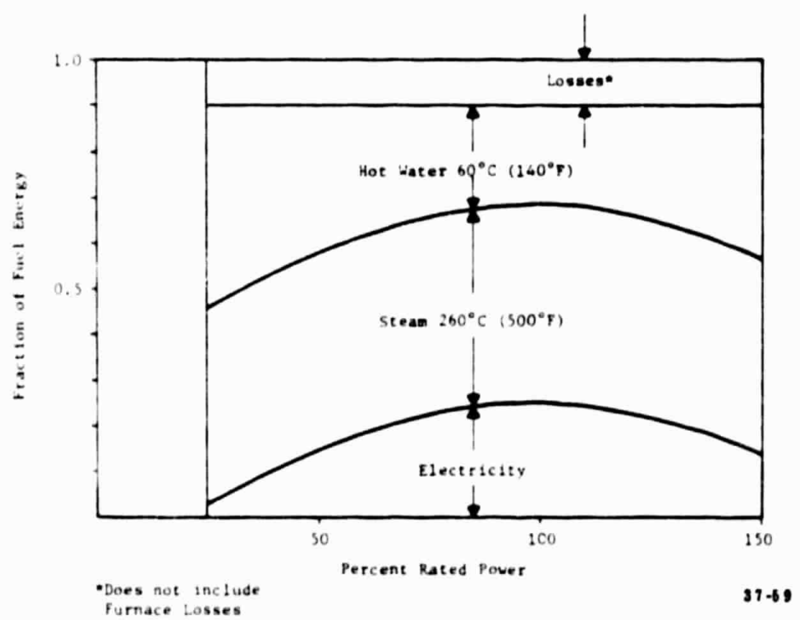


Figure III-148. Off-Design Performance Stirling Engine with Coal Atmospheric Fluidized Bed T_h = 1450°F Design Option 2

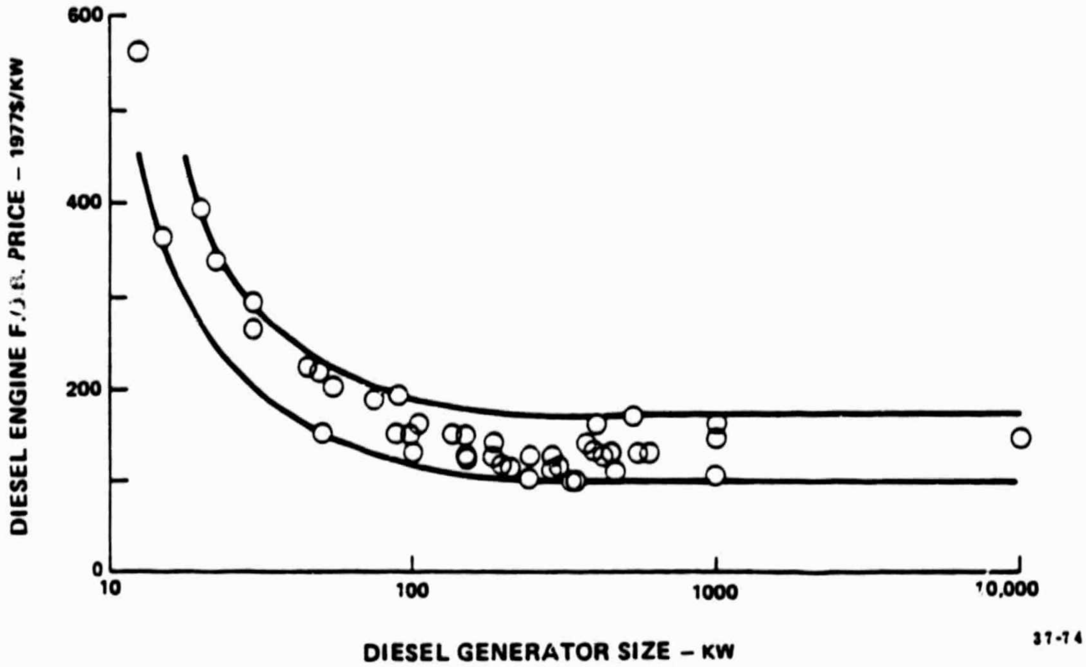


Figure III-149. Diesel Generator Prices

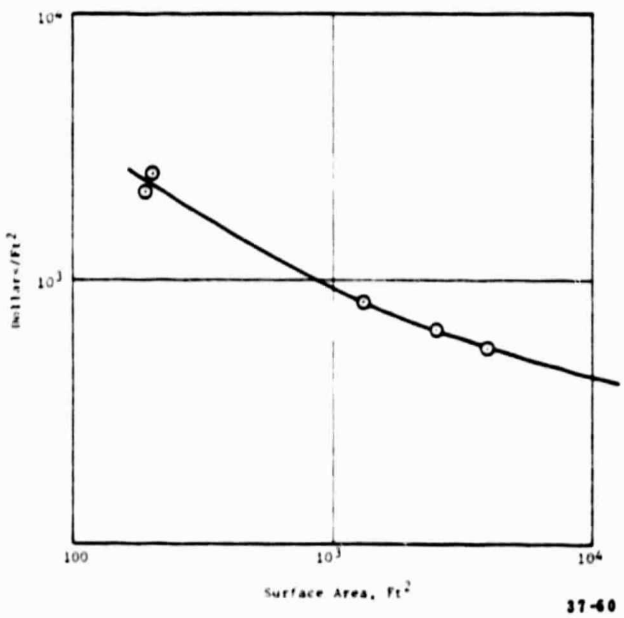


Figure III-150. Tube and Shell Heat Exchanger Costs

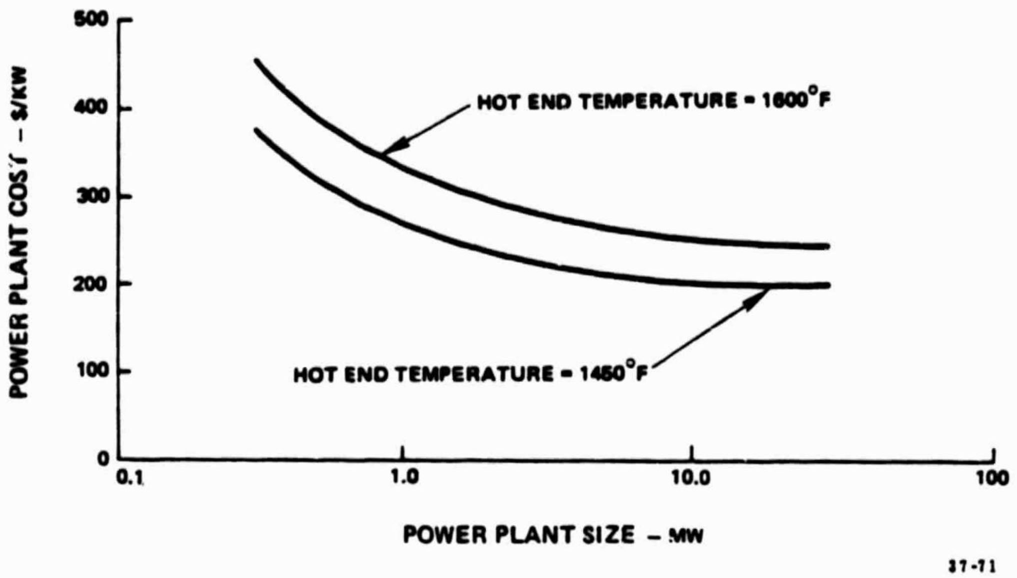


Figure III-151. Stirling Engine Power Plant Equipment Costs

THERMIONIC ENERGY CONVERSION

INTRODUCTION

The thermionic energy converter is an electronic device for converting heat directly to electric power. Unlike most heat engines, the thermionic converter uses the electrons themselves as a working fluid rather than a gas or liquid. As a result the device is simple, employing no moving parts or high pressures.

A typical thermionic converter configuration consists of a hot electrode (the emitter) facing a cooler electrode (the collector) inside a sealed enclosure containing a highly conductive low-pressure plasma. Electrons flow from the emitter to the collector across a narrow gap and deliver electric power as they return to the emitter via the external electric load. Typically the emitter is maintained between 1700 and 3150°F and the collector (or heat rejection) temperature is between 600 and 1500°F. The output of the converter is between 0.5 and 1.0 volt with a current proportional to the electrode area. Typical current densities lie between 5 and 30 A/cm^2 providing output electrical power densities between 2 and 30 W/cm^2 (2 to 30 kWe/ft^2). Higher output voltage is obtained by connecting converters in series or by using step-up transformers.

The actual converter configuration used depends strongly on the application. Converters developed to provide electrical power in space, using a nuclear reactor heat source, are small cylinders filled with nuclear fuel and operate in the core of the reactor. Solar heated converters typically have been small planar devices with the emitter electrode directly heated by the solar concentrator. For large scale industrial or power plant applications a new power module design, called a Thermionic Heat Exchanger (THX), is being evaluated by the U.S. Department of Energy.

As illustrated in Figure III-152, each THX power module is a large, cylindrically-symmetrical energy-conversion unit suitable for converting heat from any high-temperature heat source to electric power. Heat is transferred from the heat source, described in Volume IV and shown schematically in Figure III-153, to the

thermionic converters using vertical lithium-filled heat pipes 6 inches in diameter and 40 feet long. The lower, or evaporator, section of the heat pipe absorbs heat at the relatively low heat flux provided by the furnace and releases it in the upper, or condenser, section at a high heat flux to the emitters of the THX module. The heat pipes used in this study operate at only 10 percent of their theoretical capability and, therefore, are essentially isothermal, helping to ensure stable converter operation and minimizing structural problems caused by differences in thermal expansion.

Electrical output power from the THX is first transformed to higher voltage (600V) chopped DC using transformers which are an integral part of the THX module as shown in Figure III-154. This transformation is made possible by the fact that the THX module itself can be electrically switched on and off. The output power of the THX module is then further conditioned using a separate inverter system. Steam pipes thermally bonded to the collector of the module use the heat not converted to electric power to produce process steam.

The thermionic converter portion of the THX module and the output power coupling system in each THX are located outside the furnace--away from the combustion environment--allowing ready access for inspection, maintenance, and replacement. Their modular nature also serves to minimize converter development and production costs and may permit their use as retrofit units to improve the efficiency of existing systems.

In the design for the cogeneration applications, thermionic heat exchangers are installed vertically in a fossil-fueled furnace. Heat pipes serve as the inner walls of the furnace. Coal-derived boiler-fueled burners are located on two opposing walls with secondary air for emission control. Thermionic heat pipes can also be installed in the combustion chamber in a curtain arrangement

Liquid fuel was selected for this study because it permits a simpler furnace design and control system in sizes suitable for cogeneration. Fluid bed coal combustion,

which typically operates in the 1500 - 1600°F range, does not offer the high temperatures necessary for thermionic conversion. High temperature coal furnaces would require flue gas desulfurization which is very expensive in the sizes involved in this study. Therefore, liquid fuel was selected as being a reasonable choice for the thermionic conversion system which could reach commercial service in the 1985 - 2000 time period.

The THX module has a number of features that make it particularly attractive for cogeneration applications:

- The THX is modular, permitting its use in systems of any size.

- All of the heat used by the THX module is either converted to electric power or is available at high temperatures (700°F or above) suitable for almost any process requirement. Thus the fuel utilization of the THX is very high. Fuel utilization for a system using THX modules is limited only by the efficiency of the furnace (typically 88 percent). The THX modules can be used in series with a steam turbine-generator if an electrical conversion efficiency exceeding 25 percent is required.

- The efficiency of the THX module is high at part load. As an electronic device, its efficiency can be changed as needed to meet changing requirements for process heat or electrical power while maintaining constant material temperatures.

- The THX module can directly use any high temperature heat source or fuel, including coal, oil, gas, or coal-derived fuels, and nuclear or solar energy.

- The THX module has no moving parts or high pressures. It operates silently.

CONVERSION SYSTEM DESCRIPTION

This study represents the first effort to design a THX module suitable for cogeneration applications. As a result it draws heavily from the DOE program to evaluate THX modules for use in coal-fired central station power plants. To limit the range of design parameters to be considered, a number of ground rules were established. These included the basic size of the module, including the length of its heat pipe (40 feet) sized to match the furnace, and its diameter (6 inches). An emitter temperature of 2400°F was selected; no attempt was made to better match the THX modules to the system by using a combination of this and lower temperatures, a procedure that improves performance in the larger central station power plant application. Conventional furnace-to-heat-pipe heat flux of 11W/cm^2 was used, and no effort was made to increase that value, even though cost savings could result. The inductively coupled THX design was considered. Series coupling between THX modules is possible and is being considered for the central station applications. Optimizations were performed to determine the operating current density in the module (20 A/cm^2) and the number of cells in each.

Thermionic energy conversion is a new process developed originally for use in the space program. Converters developed for that application have demonstrated the performance and reliability of the basic conversion process in converter life tests exceeding 46,000 hours. The practicality of operating systems incorporating large numbers of converters has been repeatedly demonstrated in the USSR, where complete nuclear thermionic reactor systems incorporating up to 400 converters have been successfully operated. The DOE is currently conducting a program to develop power module designs for use in fossil-fuel-fired power plants, with the objective of commerical use by 1987. The program is concentrating on developing materials to protect the metal surfaces of the modules from the combustion environment (ongoing furnace life tests of such materials have reached 18,000 hours), establishing the operating characteristics of the large converters used in THX designs (THX-scale converter tests have stably driven over 6500 amperes with control typical of smaller devices) and improving converter electrical performance.

Small heat pipes have been successfully operated for more than 10,000 hours at temperatures 500-700°F above that required by a THX module. Tests of large THX-scale heat pipes are now beginning. All of these components are being brought together in a full scale THX power module test scheduled for 1981. Converters developed for the space program typically provided conversion efficiencies of up to 14 percent, but with emitter temperatures of 2950-3150°F and heat rejection (collector) temperatures of 1500°F. Recent work has reduced the peak temperature required for such performance by approximately 450°F. The program is now concentrating on using the lower terrestrial heat rejection temperatures (800°F) and relaxation of other constraints to improve efficiences to 20-25 percent by 1982 and to 35% or better by 1987. The performance level assumed in this study corresponds to that which could be available in 1985.

As was successfully done in the space program, the DOE thermionic central station power plant program uses the modular nature of the converter to permit rapid iterative development of thermionic power modules over the next ten years. The development of THX power modules suitable for cogeneration purposes could be accomplished in parallel with the central station program with benefits for both efforts.

PERFORMANCE CHARACTERISTICS

Two steps are required to specify a THX power module for cogeneration purposes. The first is to design a converter whose size and operating temperatures are optimum for a given power level. The second is the determination of the converter performance with operation off the design point.

Given the emitter temperature and heat pipe dimensions, the THX Design and Analysis Computer program was used to generate the optimum THX Design. The program models the performance and cost characteristics of a THX and allows variation along any of the component dimensions or converter temperatures, thus permitting study and optimization of the THX design.

An emitter temperature of 2400°F was chosen for the study, based on past experience. The heat pipe length was dictated by the furnace dimensions. Four converter designs were specified based on the requirement for steam at two different temperatures and two different locations in the furnace, in the middle ("curtain" or "whole" pipe) and adjacent to the wall ("half pipe"). Table III-52 lists the details of each of the four THX designs.

TABLE III-52

THERMIONIC HEAT EXCHANGER MODULE CHARACTERISTICS

DESIGN POINT

Emitter Temperature (°F)	Collector Temperature (°F)	Heat Pipe Length (ft.)	Converter Efficiency %	Power (kWe)	Cells	Location	Furnace Heat Flux (W/cm^2)
2400	1113	40	25.0	102.3	3	Curtain	11
2400	1113	40	25.3	51.7	2	Wall	11
2400	763	40	25.4	103.9	3	Curtain	11
2400	763	40	25.6	52.5	2	Wall	11

Figures III-155 through III-158 show the capital cost of a THX module as a function of output power for each of the four different designs. The figure is actually a family of curves each generated for constant N, the number of cells in the module. The optimum number of cells increases with power output because the internal resistance losses in each cell become excessive if the cells themselves become too large. Conversely, increased complexity results in increased costs if too many cells are used.

The curves show that capital costs decrease as the power of the module is increased, but the cost reduction becomes small at power greater than about 80 kWe/module.

Figure III-159 shows four envelopes taken from Figures III-156 through III-159. The two half-pipe designs are more expensive than the whole pipe designs since they receive heat into only one side and require more heat pipe length for any given output.

Figure III-161 shows capital cost as a function of emitter temperature. Above 2300°F the capital costs rise due to the thicker walls needed to sustain strength at higher temperatures. While the assumed emitter temperature of 2400°F is a bit high from a capital cost point of view, it does provide a small increase in operating efficiency.

The second step in specifying the THX module operating characteristics was to determine off-design performance. To better understand the results it is helpful to review the operating characteristics of a typical thermionic converter

Figure III-162 shows a typical current-voltage curve for a thermionic converter. The converter may be operated at any point along this curve without changing the temperature of the emitter, collector, or cesium reservoir. Only the electrical load and the heat input need be matched. Other current voltage curves are available, again at fixed emitter and collector temperatures, if the cesium reservoir temperatures are changed.

Figure III-163 illustrates a family of such performance curves taken from a test converter. It can be seen that by changing the electrical load and the cesium reservoir temperature, any desired operating point to the left of the envelope may be selected. A specific input power level is required for operation at any particular current and voltage within the operating region.

Figures III-164 through III-167 show the off-design performance for each of the four THX modules designed for CTAS, assuming the system allows variation in both cesium temperature and electrical load resistance. Here, however, heat input to the module is shown as a function of electric power output for various values of output current and voltage. There is a vertical offset to the curves, reflecting

the fact that at low current densities radiative heat transfer is still taking place. The lines become dotted outside of the permissible operating range of the module.

An example may serve to clarify the use of a performance map. The design-operating point of the THX module characterized in Figure III-164 occurs at an output power of 102 kWe, an input power of 409 kWth, and an emitter current density of 20 A/cm^2. By changing the cesium pressure, as shown in Figure III-163, the output voltage and power of the module can be reduced, even though the electrical load is adjusted so as to maintain an emitter current density of 20 A/cm^2. If this process were continued to its extreme, the electrical output of the module would continue to drop, along the 20 A/cm^2 line in the figure, until the electrical load was in fact a short circuit and no electrical power was delivered. The input power required under these conditions would be 355 kWth, all of which would be delivered as process heat. To continue the example, the current produced by the converter could then be reduced by lowering the cesium pressure. The input heat from the furnace would be reduced simultaneously in order to maintain a constant emitter temperature. As a result the process heat provided by the module could be reduced to as little as 80 kWth before it would become necessary to reduce the heat pipe temperature. This process corresponds to moving down the left axis of the plot in Figure III-164 from 390 kWt, 20 A/cm^2 to 80 kWt.

The versatility reflected in these performance maps allows the power modules to be operated in several modes. For example, they can be controlled to operate along lines of constant output power, along lines of constant process heat production, along lines of constant input power, or along lines of constant load resistance.

For purposes of the CTAS study the simplest control scheme was assumed, that of operation at fixed cesium reservoir temperature and constant emitter temperature with varying electrical load. The part-load efficiencies of each of the four module designs, using this control system, are shown in Figure III-168.

In all of the examples above it has been assumed that the furnace is being controlled in such a way so as to maintain a constant emitter-heat pipe temperature. Of course, relieving this constraint would permit further operational flexibility at the cost of increased complexity.

A reference heat source design was selected and reported in Volume IV. This thermionic furnace consumed 141 million BTU/hour and operated at 88 percent efficiency.

The performance and cost characteristics of the thermionic converter heat source are presented in Volume IV. This heat source is included schematically in Figure III-169. Temperatures, pressures, and flow rates at various locations in the system are presented in Table III-56 for the design conditions.

The preheater exhaust temperature is 2200°F. This is well above the 1400°F metal heat exchanger capability projected for 1985-2000. Therefore, the preheater heat exchanger is in two sections: ceramic and metal. Ceramic heat exchangers are in the research and development stage and are projected to be capable of operation to 2200°F although the costs of a commercial ceramic heat exchanger in 1985-2000 are expected to be high.

Because of the high operating temperature of the converters, the furnace exhaust gases are 2650°F. After preheating the combustion air, there is sufficient energy in the furnace exhaust gas to generate additional steam at 600 psig and 700°F for use in the industrial process. The exhaust enters the stack at 300°F.

While the overall efficiency of the furnace does not vary significantly with size, the number of thermionic heat pipes in the walls in relation to the curtain converters will vary. Figure III-170 presents this design variation based on Figure IV-27, Volume IV. Seventeen percent of the furnace thermal output is delivered in the form of steam from the steam generator shown in Figure III-169.

The system, shown in Figure III-169, produces 7.21 megawatts electrical energy at the design point. This includes the effects of inverter efficiency of 94 percent and parasitic losses of 163 kilowatts. The furnace contains 86 wall thermionic heat pipes and 31 curtain units. The furnace efficiency is predicted to be 88.2

percent. The electrical efficiency is 17.2 percent, and the thermal energy represents 69.6 percent of the heating value of the fuel. The overall fuel utilization is 86.8 percent. The design performance is not sensitive to system size as shown in Figure III-171. Larger power plants have the same performance as the 30 megawatt size equipment.

Since the ceramic heat exchanger typically represents over one third of the cost of the heat source, an alternate heat source design without the ceramic heat exchanger was designed and reported in Volume IV. In this modification, shown in Figure III-172, the furnace exhaust gases first generate steam and then preheat the combustion air to 1400°F in a metallic heat exchanger. The temperatures, pressures, and flow rates at various locations in the system are presented in Table III-57 for the design conditions.

The design configuration without the ceramic heat exchanger produces 5.24 megawatts electric output. The furnace efficiency, parasitic requirements, and inverter performance are the same as the first design. Since the preheat temperature is reduced, the combustion air must be heated from 1400 to 2200°F in the combustion chamber rather than in the preheater. As a result, for the same inlet fuel flow, the energy reaching the converters is reduced from 104 to 77 million BTU/hour. The number of wall thermionic heat pipes is reduced to 68, and the number of curtain units is reduced to 21. The electrical efficiency is 12.3 percent, and the useful thermal energy is 74.5 percent of the higher heating value of the fuel. The overall fuel utilization is 87 percent. The performance variation with output rating is essentially constant, as in the first design.

The characteristics of the thermionic heat pipes and the furnace lead to a maximum electrical to thermal energy ratio of 0.25 with the ceramic heat exchanger. (With the metal heat exchanger, this ratio drops to 0.17). To provide higher electric output the collector temperature was raised to 1110°F, and the steam pressure and temperature were raised to 1800 psi and 1050°F. The effect on the converter performance was minimal. Passing the steam through an extraction turbine is an effective way to increase the electrical output.

Returning to a case with a ceramic preheater, Figure III-173 presents a schematic diagram of the compound steam turbine-thermionic converter energy conversion system, and Table III-58 presents temperatures, pressures, and flow rates at the design point from this system. With 86 wall and 31 curtain thermionic heat pipes, the converter output electrical energy is only reduced slightly to 7.09 megawatts with the higher collector temperature (compared to 7.21 megawatts). However, a single extraction steam turbine provides 4.99 megawatts and steam at 600 psig. The total electrical output is 12.08 megawatts. The electrical efficiency is 28.2 percent and the thermal energy represents 30.5 percent of the higher heating value of the fuel. Figure III-174 presents the variation in electrical and thermal output with design size. This configuration emphasizes electrical output since only 50 percent of the steam was extracted from the turbine for process use. Higher extraction flows would reduce the electrical and enhance the thermal output of the system. Reducing the extraction flow to zero and sending all of the steam through the turbine to the condenser would provide maximum electrical output with no thermal energy. In this limiting case, the electrical efficiency would be 36 percent based on the higher heating value of the fuel.

Another design option was developed using the metallic heat exchanger and the compound steam turbine-thermionic converter system, Figure III-175. The conditions at various points within the system are listed in Table III-59. In this case, the electrical output was 8.19 megawatts of which 3.03 megawatts were developed by the steam-turbine-generator. Higher overall fuel utilization was achieved with 80 percent of the steam turbine flow being extracted at 600 psig. The electrical efficiency is 19.5 percent, and the useful thermal output is 52.0 percent giving an overall fuel utilization of 71.5 percent at the design conditions. The variation of the performance of this system with power plant size is presented in Figure III-176.

The last thermionic compound system design option incorporates a metallic heat exchanger and an extraction turbine. The design is the same as the previous option, Figure III-175, with the exception that steam is extracted from the turbine at 50 psig instead of 600 psig. The turbine electrical output is raised to 5.8

megawatts (from 3.0 megawatts) and the resulting overall output is 11.0 megawatts at the design point. Figure III-177 indicates the variation in system performance with power plant size.

Table III-60 summarizes the principal design parameters for the thermionic conversion system design options.

ESTIMATED COSTS

The cost of the thermionic heat exchanger units was estimated based upon material quotations and labor estimates. The resulting cost and weight estimates are presented in Table III-61. The inverter characteristics are included in Table III-62.

The complete energy conversion system consists of the thermionic heat pipes, the furnaces and associated equipment, the inverter and the balance of plant. In the analysis described in Volume V, the heat source and balance of plant costs from Volume IV are combined with the thermionic heat exchanger and inverter costs to define the system costs. The cost data presented here are limited to the thermionic heat pipes, inverters, and steam turbine-generator-condensers, as appropriate. The steam turbine costs are discussed further in the steam turbine section of this volume. The installation costs are based upon the balance of plant item covering installation of erected equipment in Volume IV.

The estimated costs for the thermionic heat pipes and the inverter are included in Figure III-178 conversion system number 34. While the performance of these systems does not vary significantly with powerplant size, there is some variation in estimated cost because of the increase in number of wall units in respect to curtain units as the size is reduced. The estimated costs of the components of the compound thermionic conversion systems) conversion system number 35) are presented in Figures III-179, III-180, and III-181.

OPERATION AND MAINTENANCE

The THX modules are assumed to have an operating life of 100,000 hours. This is consistent with the operating lifetimes achieved by large power tubes. The NASA objective for nuclear reactor-thermionic space power systems using lithium-molybdenum heat pipes is 70,000 hours; however, temperatures in that design are hotter than in the cogeneration systems. Similarly the longest converter test to date provided stable performance for 46,000 hours with an emitter temperature 660°F above those in the cogeneration system design. The THX modules themselves have been designed to last 30 years (263,000 hours) prior to failure from creep to rupture, thus providing a conservative safety margin.

Replacement of the entire THX module at full cost is assumed after 100,000 hours. The cost of removal and reinstallation is estimated to be 10% of the capital cost of the units. This is twice the original installation cost defined by Bechtel National. The O&M cost was based on a 24-year plant life and 7500 hour/year duty; no credit was assumed for material or component salvage.

Inventory requirements for maintenance purposes will be dominated by a requirement for back-up THX manpower modules. An inventory requirement equivalent to 3% of the total number of THX modules in the furnace was assumed. The inventory cost was defined to be the cost of the money invested at a rate of 5-1/2%. The estimated operating and maintenance charges (excluding fuel) are presented in Table III-60.

TABLE III-60

THERMIONIC CONVERTER ESTIMATED MAINTENANCE COSTS

Collector Temperature °F	Type	Capital Cost $\$/kW_e$	Replacement $mills/kW_H$	Inventory $mills/kW_H$	Total $mills/kW_H$
1113	Curtain	316	1.55	.06	1.61
763	Curtain	310	1.51	.06	1.58
1113	Wall	469	2.29	.09	2.39
763	Wall	461	2.25	.09	2.35

PHYSICAL SIZE

The Thermionic heat exchangers are installed within the furnace. The physical or space requirements of the furnace and other balance-of-plant equipment are presented in Volume IV. The floor area required for the inverter is 0.1 square foot per kilowatt of inverter output. The floor area required for the combination of the turbine-generator and the inverter for the compound thermionic conversion systems is presented in Figure III-182 in relation to the total electrical output of the complete system.

COGENERATION APPLICABILITY

Thermionic energy conversion systems are particularly well suited to cogeneration situations requiring large amounts of high temperature process heat. The compound configuration is appropriate for industrial processes requiring moderate amounts of electricity. This study utilized a steam turbine for compound applications because the thermionic converter heat rejection temperature can be well above the temperature for organic fluids.

Heavy oil or coal derived oil serve as the basis for this study. The furnace can be modified to operate with a wide variety of light or distillate oils or gaseous fuels in addition to boiler grade oils. With a suitable flue gas desulfurizer, coal could be used directly.

The design approach using 40-foot long heat pipes suggests minimum size energy conversion systems of about one megawatt electric. Thermionic heat pipes of other sizes could provide smaller electric output, if desired.

FUTURE DEVELOPMENTS

The following technical developments are needed to bring thermionic energy converters to commercial cogeneration applications in the 1985-2000 time period:

1. Engineering development of the thermionic heat exchanger (THX) to achieve required levels of performance, capital cost, durability, and reliability.

2. Development of high temperature full scale heat pipes capable of operating in the combustion environment.

3. Development of converter electrical and control systems to produce chopped direct current converter output.

4. Development of inverters to operate with chopped direct current.

5. Development of high temperature furnace.

6. Development of high temperature heat exchangers.

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1. G. Carnasciali and G. O. Fitzpatrick, Foster Wheeler Development Corp., Livingston, N. J.; E. J. Britt, Rasor Associates Inc., Sunnyvale, Calif., "Performance and Cost Evaluation for a Thermionic Topping Power Plant"; ASME 77-WA/Ener-7.

2. Pietsch, Antony; "Coal Fired Prototype High Temperature Continuous Flow Heat Exchanger"; EPRI-AF 684 Project 545-1; AiResearch Manufacturing Company of Arizona, Division of Garrett Corp.; February 1978.

3. D. H. Brown, B. D. Pomeroy, R. P. Shah; Energy Conversion Alternatives Study, General Electric Phase II Final Report, Volume II, Advanced Energy Conversion Systems Conceptual Designs; NASA CR 134949, General Electric Company.

TABLE III-53
THERMIONIC ENERGY CONVERSION SYSTEM
CERAMIC PREHEATER
DESIGN CONDITIONS

LOCATION Figure 1	FLOW Pounds/Hour	TEMPERATURE °F	PRESSURE PSIA
1	120,700	59	15.2
2	120,700	300	15.1
3	120,700	2200	15.0
4	129,000	2650	14.8
5	129,000	1100	14.7
6	129,000	500	14.6
7	129,000	300	14.5
8	19,500	250	30.0
9	18,600	700	615.0
10	8,300	120	140.0
11		2400	
12		763	
13	26,300	700	615.0

TABLE III-54
METAL PREHEATER
DESIGN CONDITIONS

LOCATION Figure 1	FLOW Pounds/Hour	TEMPERATURE °F	PRESSURE PSIA
1	129,700	59	15.2
2	120,700	420	15.1
3	120,700	1400	15.0
4	129,000	2650	14.9
5	129,000	1700	14.8
6	129,000	890	14.7
7	129,000	580	14.6
8	129,000	300	14.5
9	39,200	700	615.0
10	8,300	120	140.0
11		2400	
12		763	
13	46,400	700	615.0

TABLE III-55
COMPOUND THERMIONIC ENERGY CONVERSION SYSTEM
CERAMIC PREHEATER
DESIGN CONDITIONS

LOCATION Figure 1	FLOW Pounds/Hour	TEMPERATURE °F	PRESSURE PSIA
1	120,700	59	15.2
2	120,700	300	15.1
3	120,700	2200	15.0
4	129,000	2650	14.8
5	129,000	1100	14.7
6	129,000	500	14.6
7	129,000	300	14.5
8	19,500	250	30.0
9	15,000	1050	1815.0
10	8,300	120	140.0
11		2400	
12		763	
13	55,400	1050	1815.0
14	35,200		615.0
15	35,200		3.0

TABLE III-56
COMPOUND THERMIONIC ENERGY CONVERSION SYSTEM
METAL PREHEATER
DESIGN CONDITIONS

LOCATION Figure 1	FLOW Pounds/Hour	TEMPERATURE °F	PRESSURE PSIA
1	120,700	59	15.2
2	120,700	420	15.1
3	120,700	1400	15.0
4	129,000	2650	14.9
5	129,000	1700	14.8
6	129,000	890	14.7
7	129,000	580	14.6
8	129,000	300	14.5
9	34,300	1050	1815.0
10	8,300	120	140.0
11		2400	
12		763	
13	40,600	1050	1815.0
14	59,900		615.0
15	15,000		3.0

TABLE III-57
THERMIONIC ENERGY CONVERSION
SYSTEM DESIGN VARIABLES

Conversion System	34	34	35	35	35
Design Option	1	2	1	2	3
Steam Turbine-Generator	No	No	Yes	Yes	Yes
Heat Exchanger	Ceramic	Metal	Ceramic	Metal	Metal
Wall Thermionic Heat Pipes	86	68	86	68	68
Curtain Thermionic Heat Pipes	31	21	31	21	21
Collector Temperature - F	763	763	1110	1110	1110
Converter Efficiency - percent	25.5	25.5	25.2	25.2	25.2
Extraction Pressure - psig	--	--	600	600	50
Extraction - Percent	--	--	50	80	80
Furnace Efficiency - percent	88.2	88.2	88.2	88.2	88.2
Inverter Efficiency - percent	94	94	94	94	94
Electrical Efficiency - percent	17.4	12.3	28.2	19.5	26.2
Thermal Output - percent	70.0	74.5	30.5	52.0	46.0

TABLE III-58

THERMIONIC HEAT EXCHANGER CHARACTERISTICS

Location in furnace	Wall		Curtain	
Heat Pipe Length - feet	40		40	
Number of cells	2		3	
Emitter Temperature °F	2400		2400	
Collector Temperature °F	760	1110	760	1110
Weight - pounds	2660	2678	4632	4669
Output - KW	52.5	51.7	103.9	102.3
Estimated Cost	$24,200	$24,250	$32,200	$32,300

TABLE III-59
INVERTER CHARACTERISTICS

Efficiency	94%
Specific Cost	$50/KW
Specific Weight	10 lbs/KW
Specific Floor Area	0.1 Sq.Ft./KW

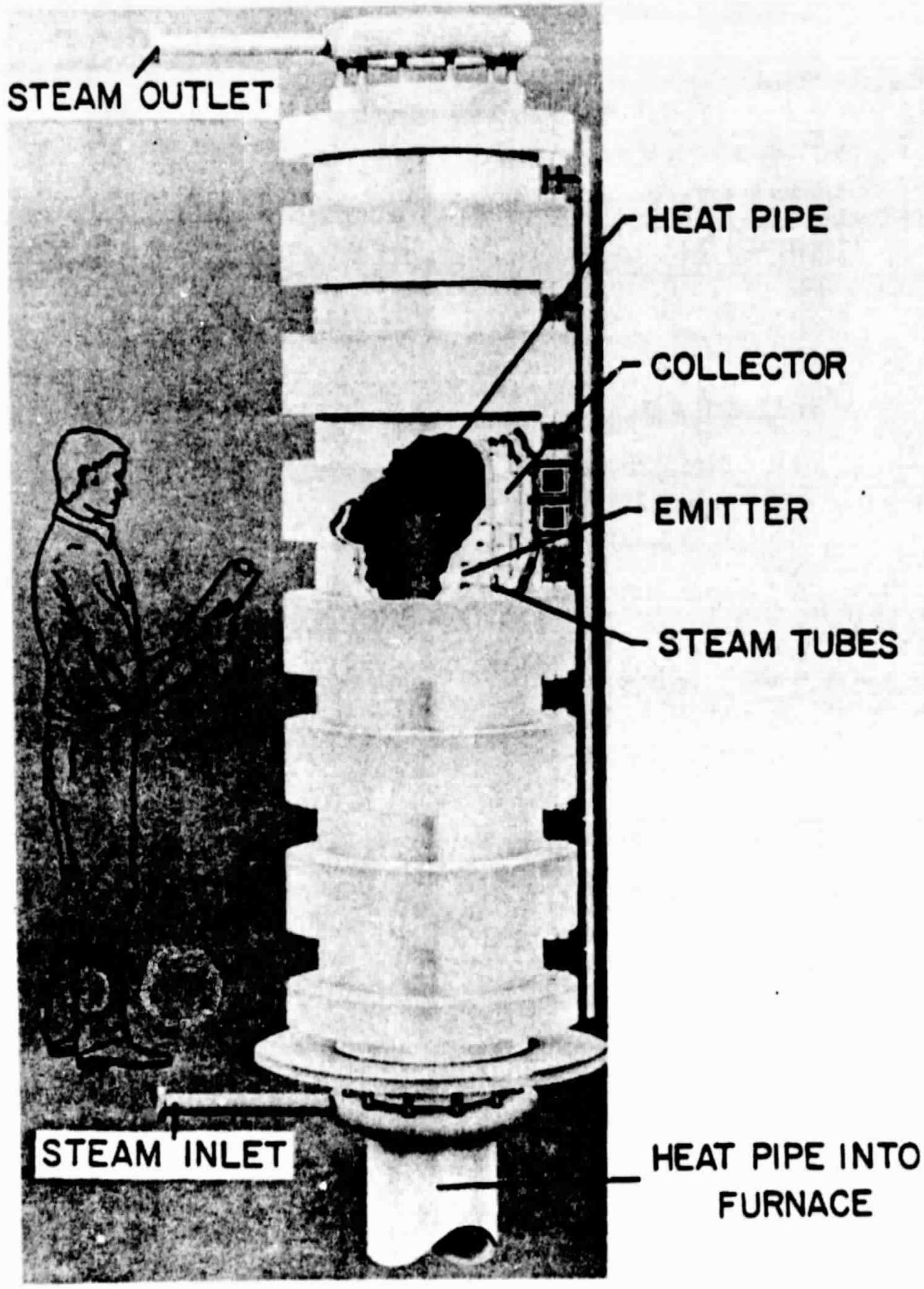

Figure III-152. Thermionic Heat Exchanger

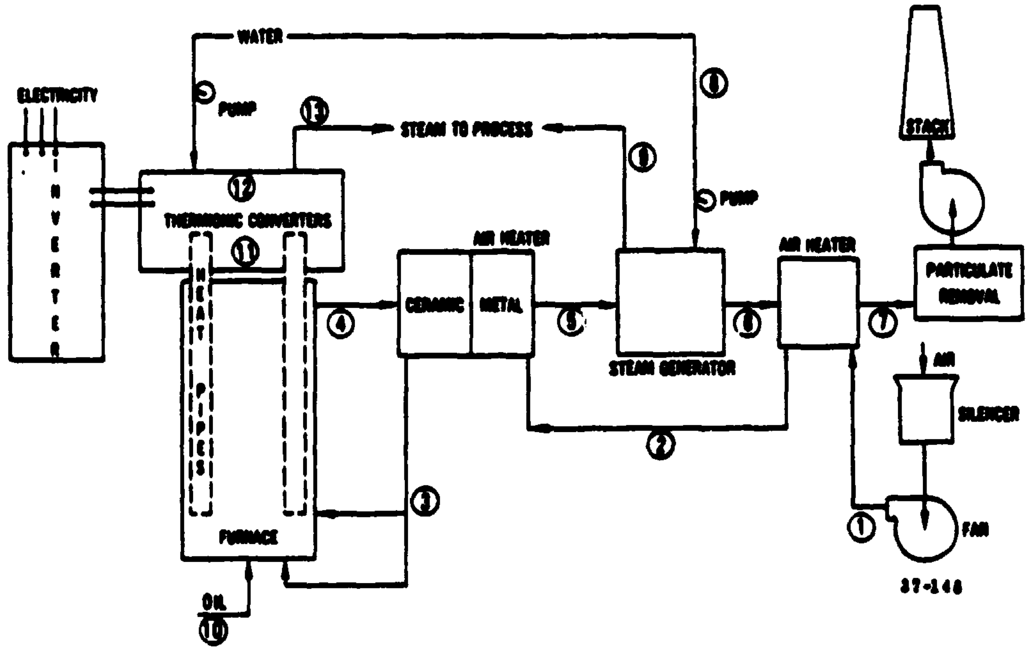

Figure III-153. Thermionic Energy Conversion System Schematic Diagram - Coal-Derived Boiler Fuel - Ceramic Preheater

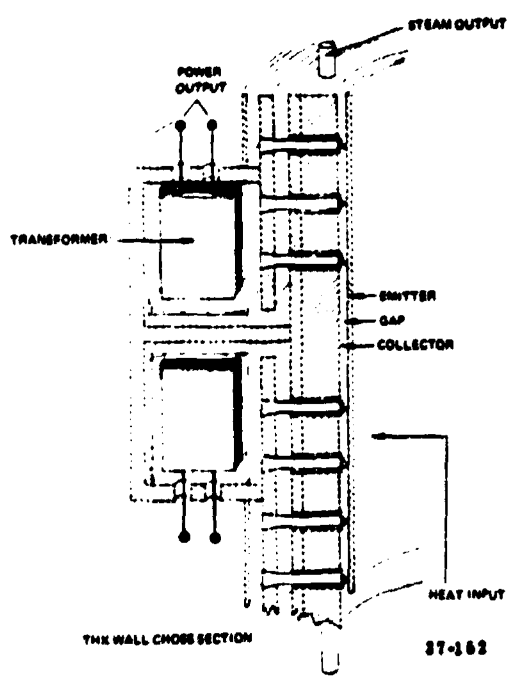

Figure III-154. Thermionic Heat Exchanger Cross Section

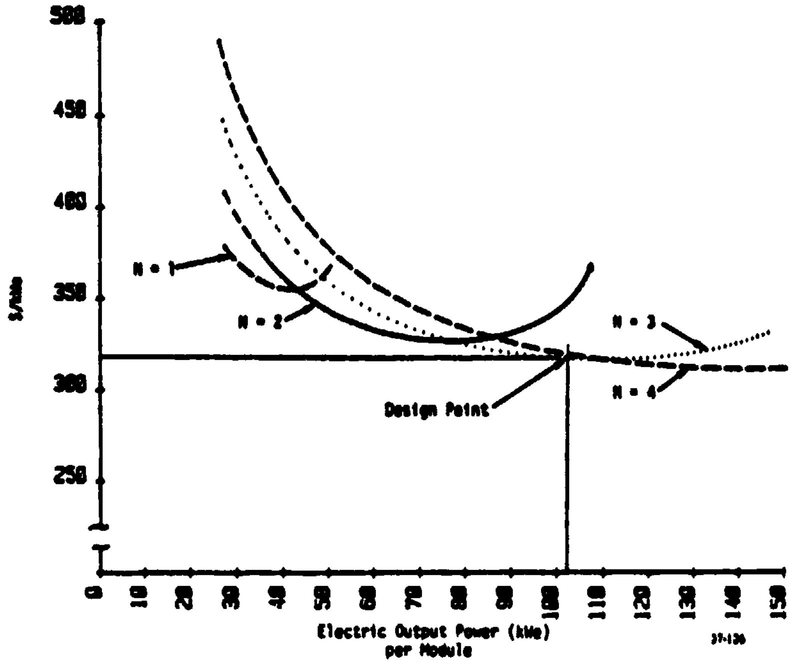

Figure III-155. Parametric THX Converter Cost Estimates - Whole Pipe - Collector Temperature 1110°F

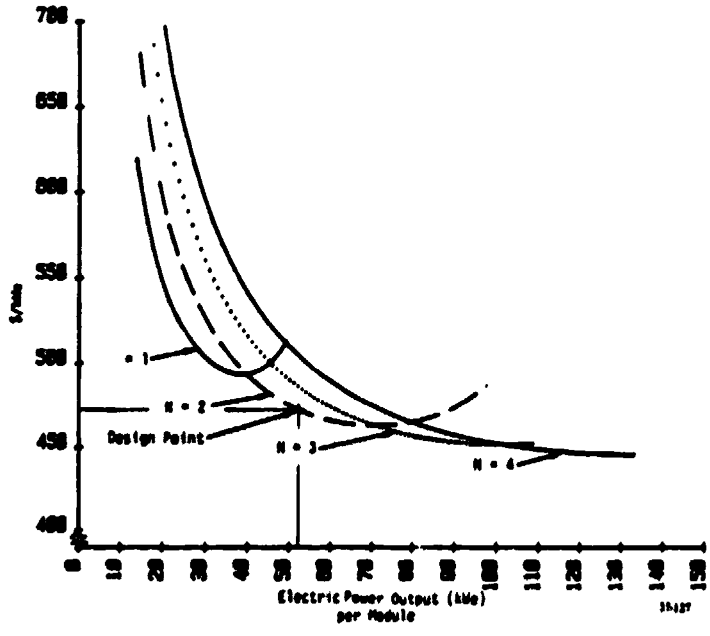

Figure III-156. Parametric THX Converter Cost Estimates - Half Pipe - Collector Temperature - 1110°F

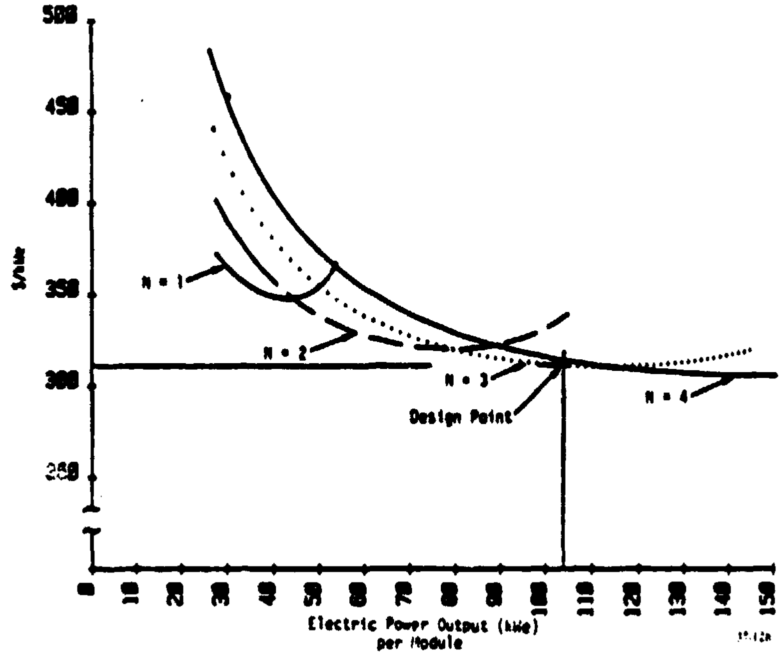

Figure III-157. Parametric THX Converter Cost Estimates - Whole Pipe - Collector Temperature - 760°F

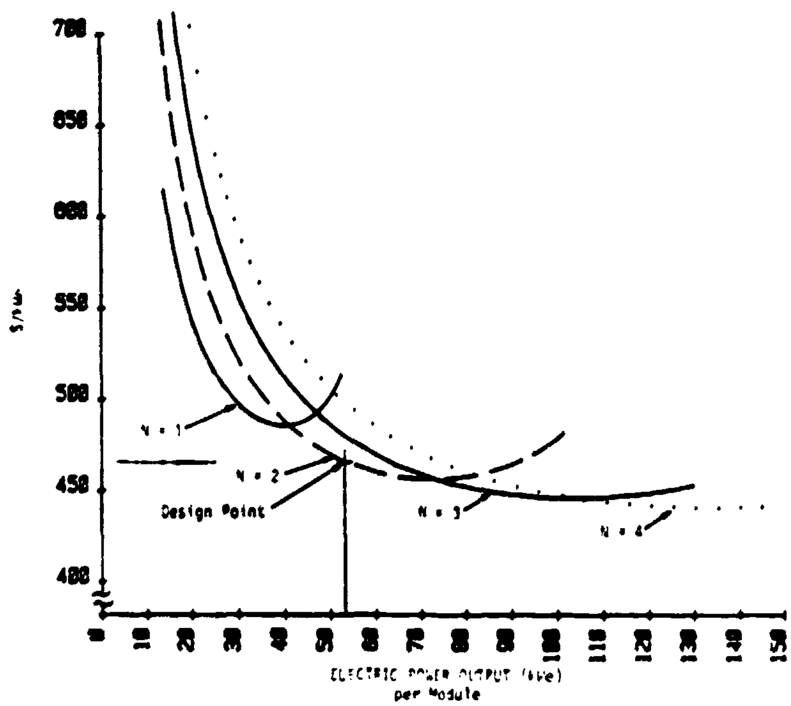

Figure III-158. Parametric THX Converter Cost Estimates - Half Pipe - Collector Temperature - 760°F

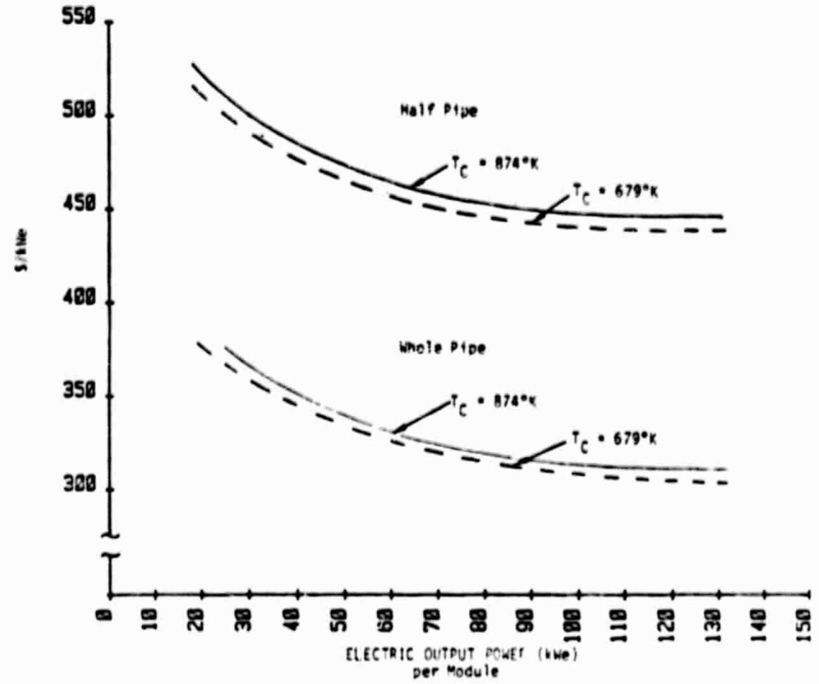

Figure III-159. Envelope of THX Converter Parametric Cost Estimates - Emitter Temperature - 2400°F

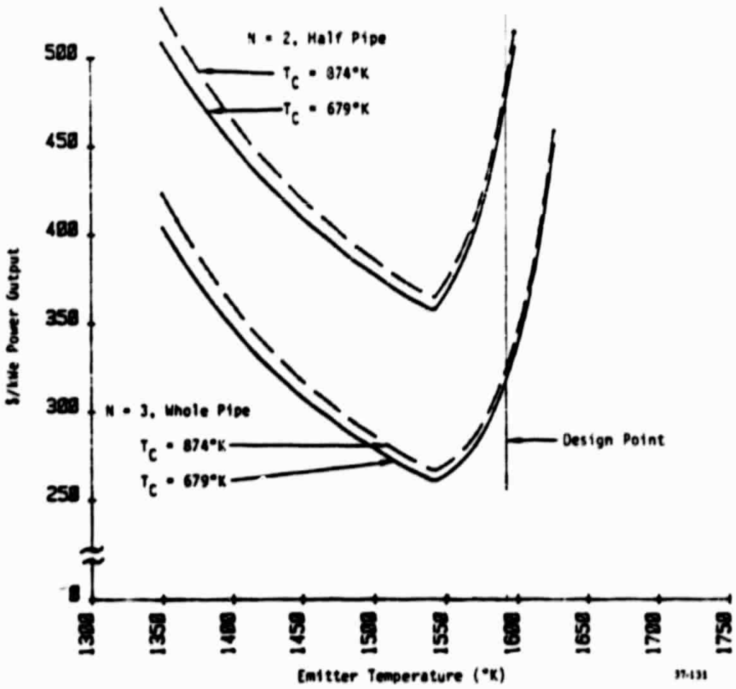

Figure III-160. THX Converter Parametric Cost Variation with Emitter Temperature

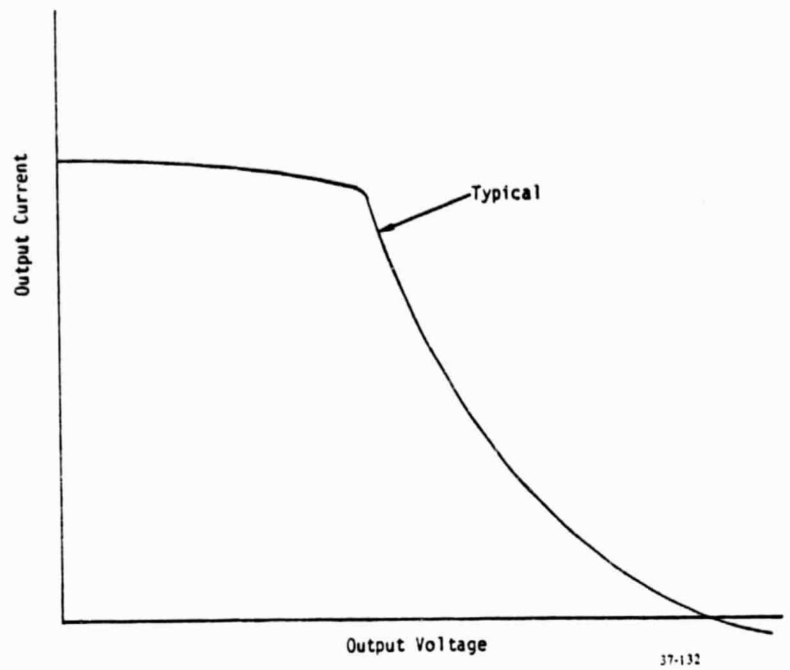

Figure III-161. Typical THX Converter Current Voltage Relation

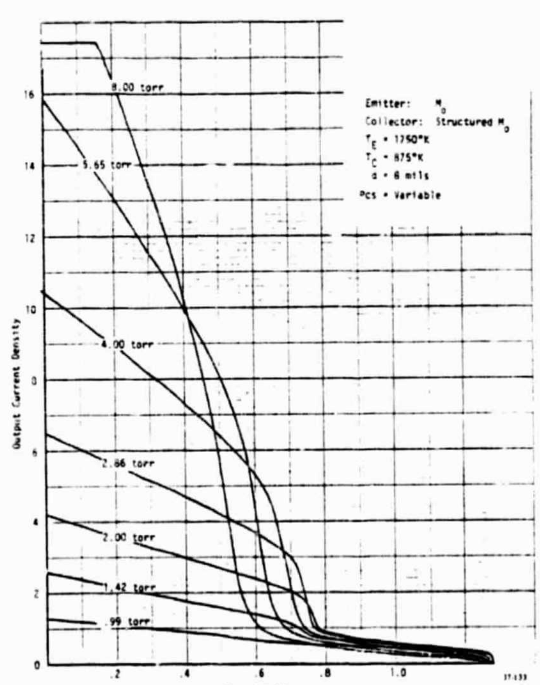

Figure III-162. Performance Curves From A Test Converter

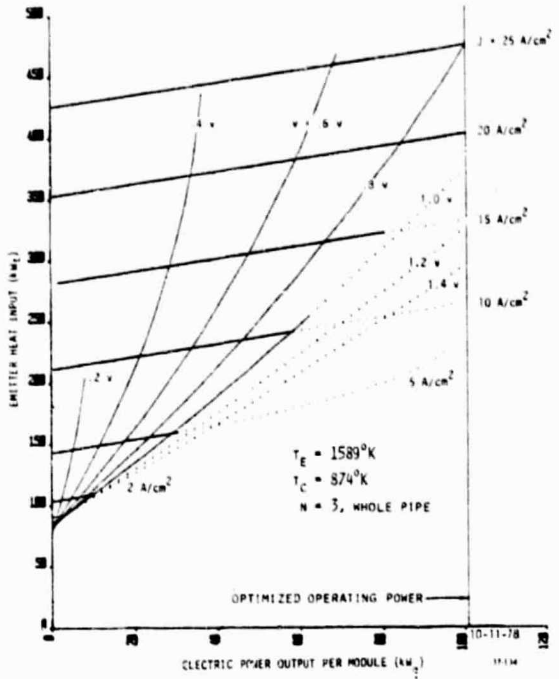

Figure III-163. THX Converter Off-Design Performance - Whole Pipe - Collector Temperature - 1110°F

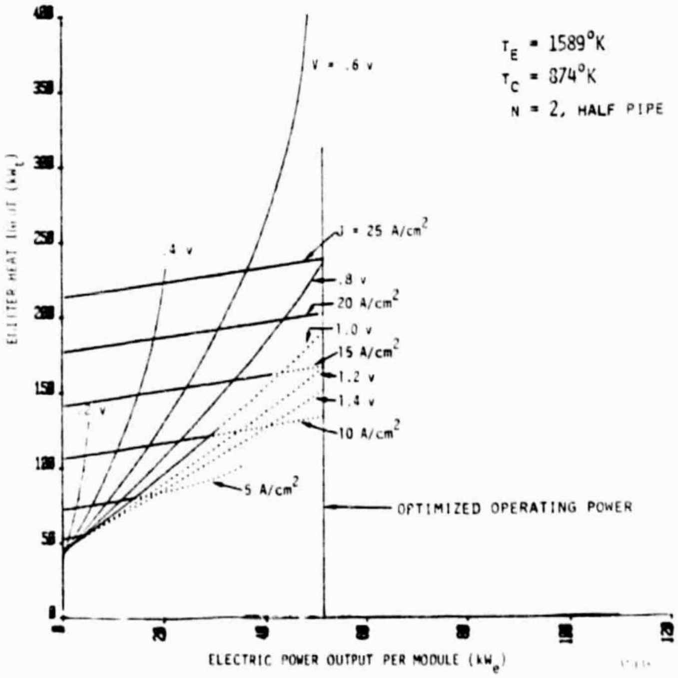

Figure III-164. THX Converter Off-Design Performance - Half Pipe - Collector Temperature - 1110°F

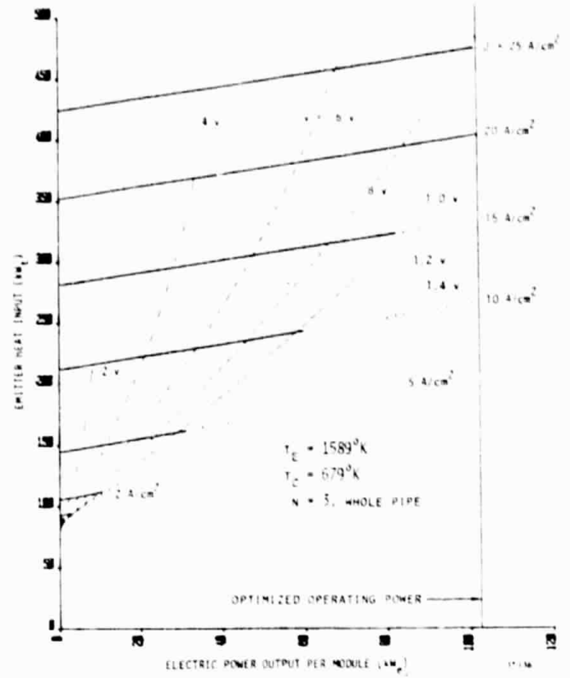

Figure III-165. THX Converter Off-Design Performance - Whole Pipe - Collector Temperature - 760°F

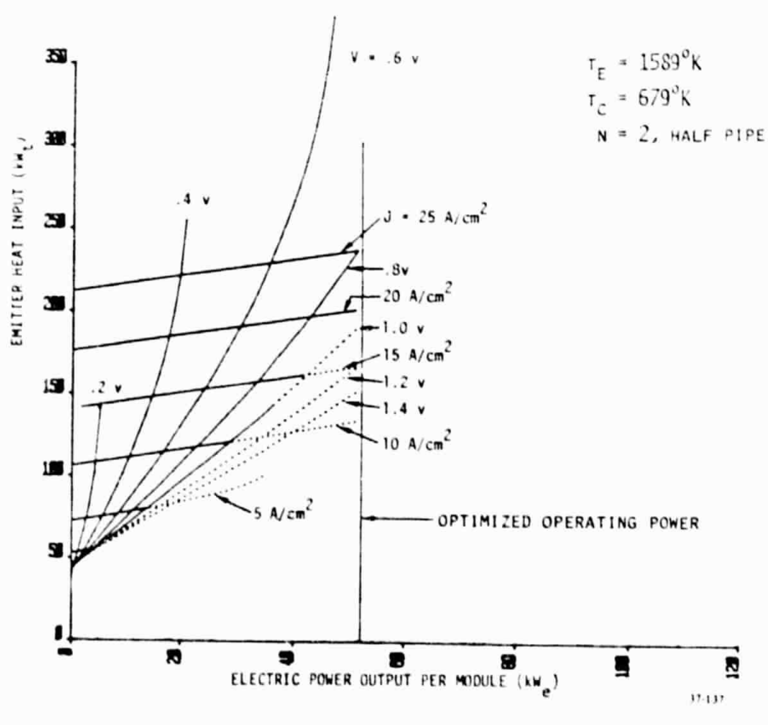

Figure III-166. THX Converter Off-Design Performance - Half Pipe - Collector Temperature - 760°F

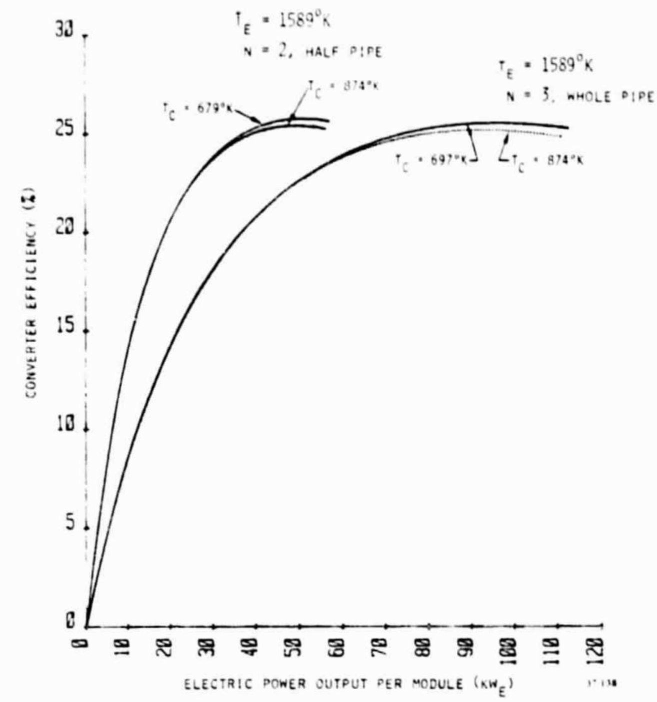

Figure III-167. THX Converter Part Load Performance with Constant Emitter and Cesium Reservoir Temperature

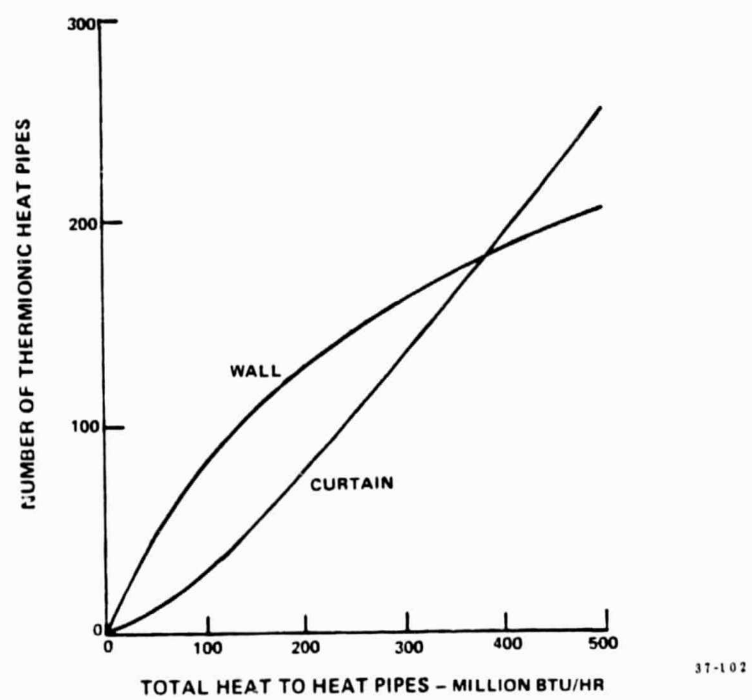

Figure III-168. Thermionic Conversion System Heat Pipe Selection

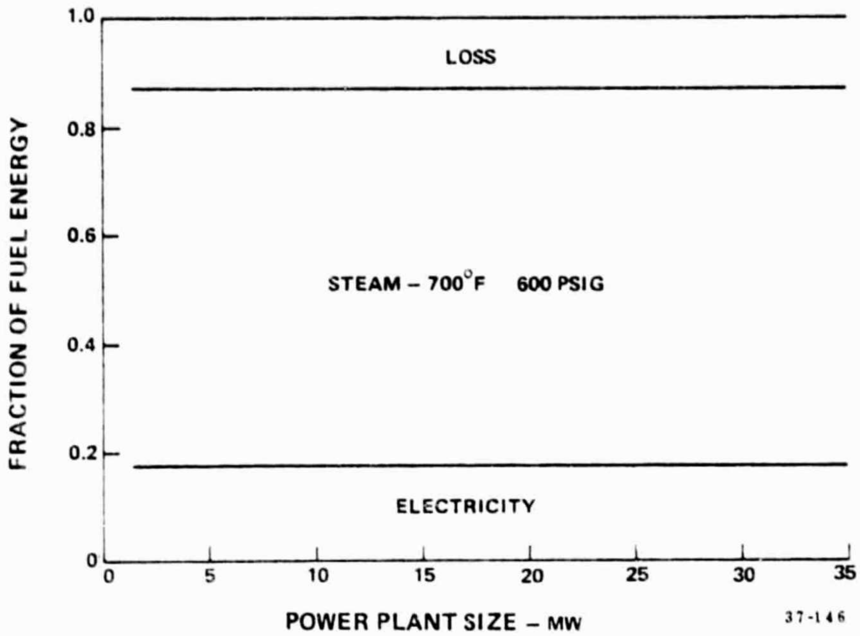

Figure III-169. Thermionic Conversion System Performance - Coal-Derived Residual Oil - Ceramic Preheater - Design Option 1

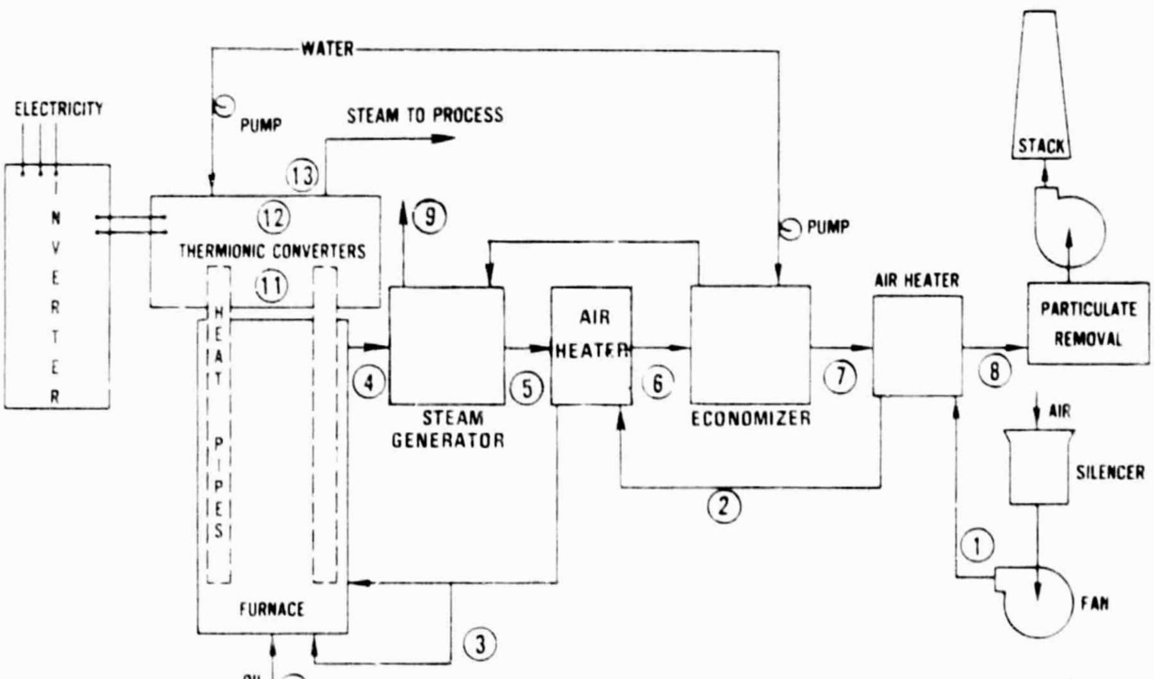

Figure III-170. Thermionic Energy Conversion System Schematic Diagram - Coal-Derived Boiler Fuel - Metal Preheater

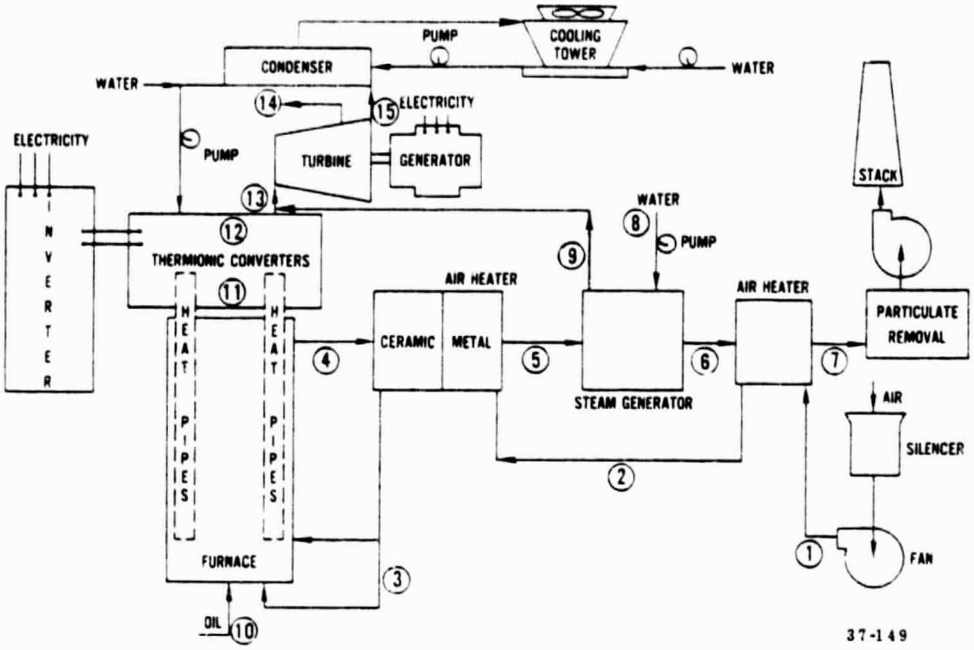

Figure III-171. Compound Thermionic Energy Conversion System Schematic Diagram - Coal-Derived Boiler Fuel - Ceramic Preheater

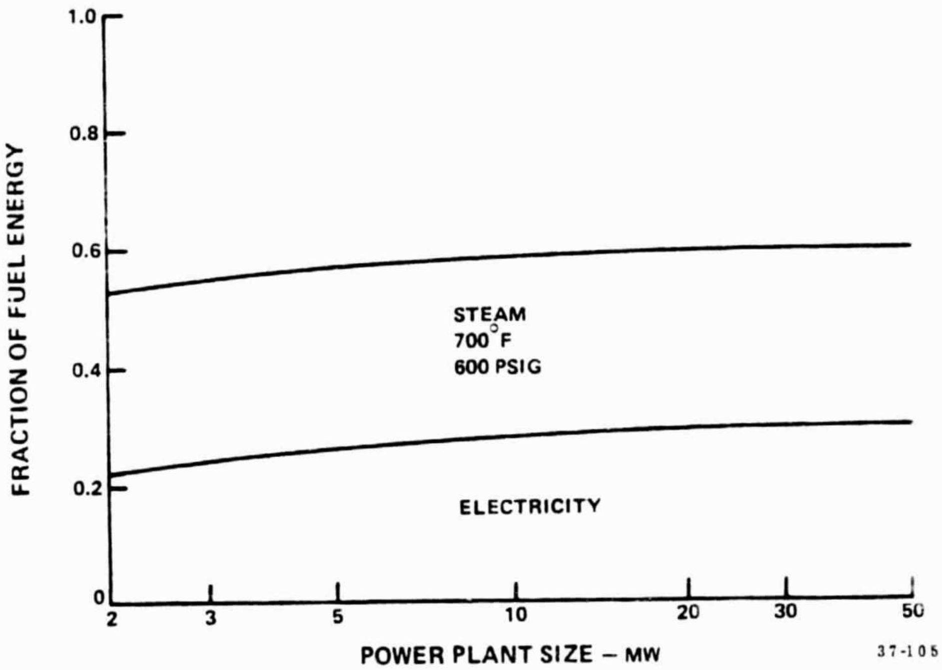

Figure III-172. Compound Thermionic Conversion System Performance - Coal-Derived Boiler Fuel - Ceramic Preheater - Design Option 1

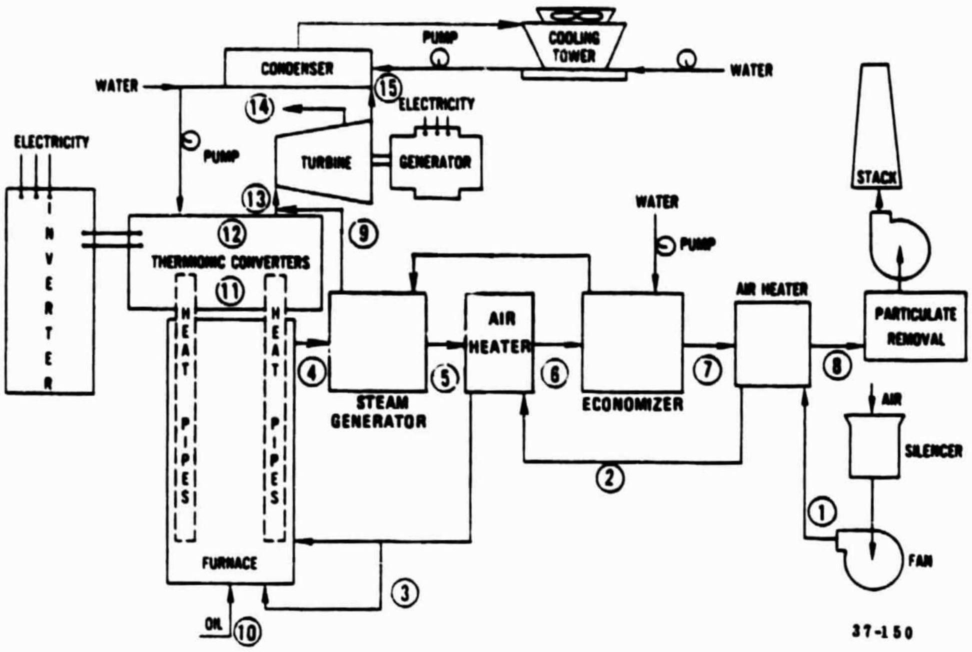

Figure III-173. Compound Thermionic Energy Conversion System Schematic Diagram - Coal-Derived Boiler Fuel - Metal Preheater

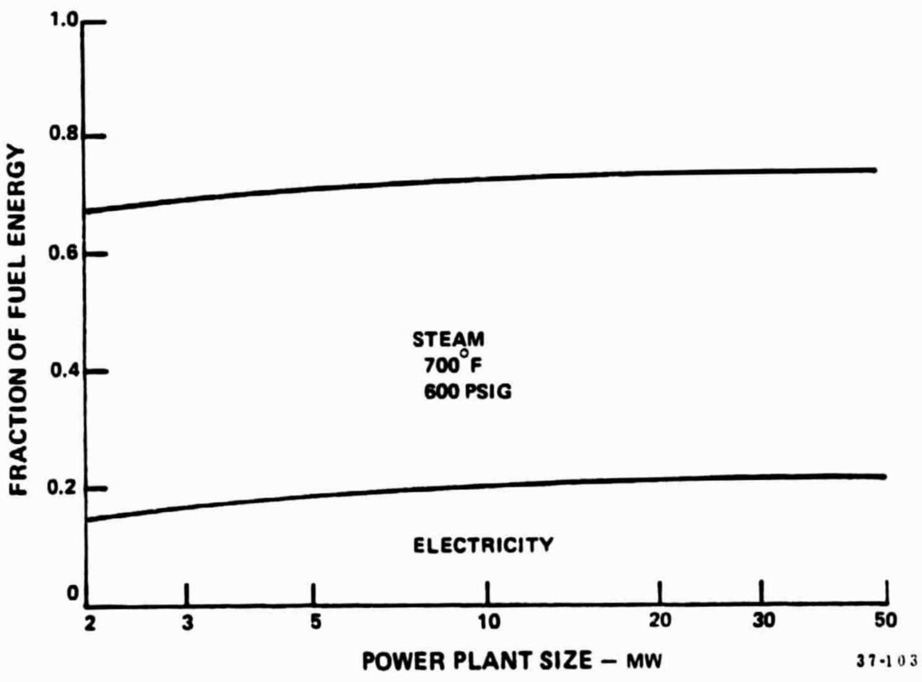

Figure III-174. Compound Thermionic Conversion System Performance - Coal-Derived Boiler Fuel - Metal Preheater - Design Option 2

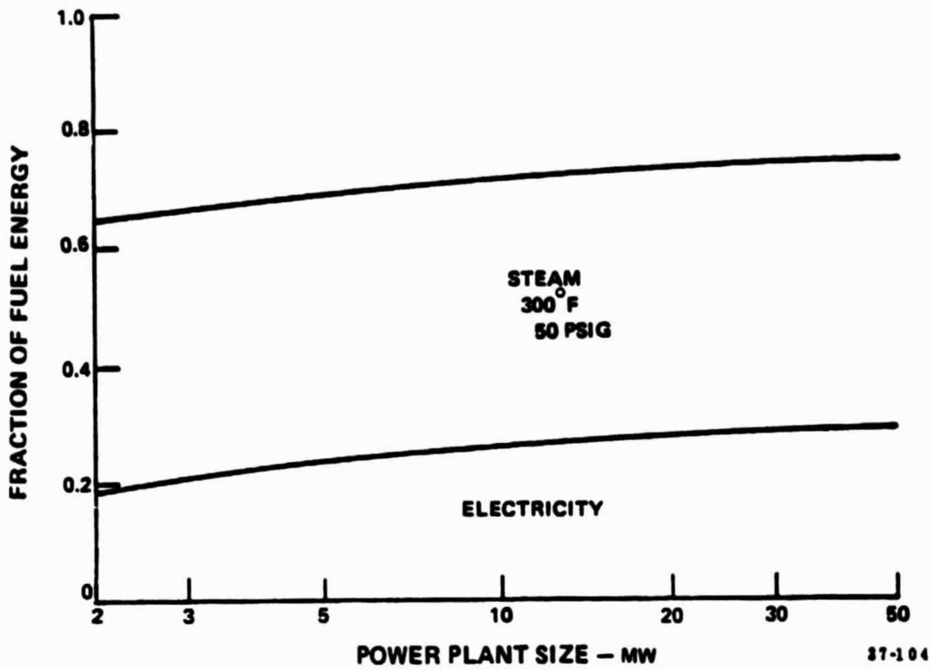

Figure III-175. Compound Thermionic Conversion System Performance - Coal-Derived Boiler Fuel - Metal Preheater - Design Option 3

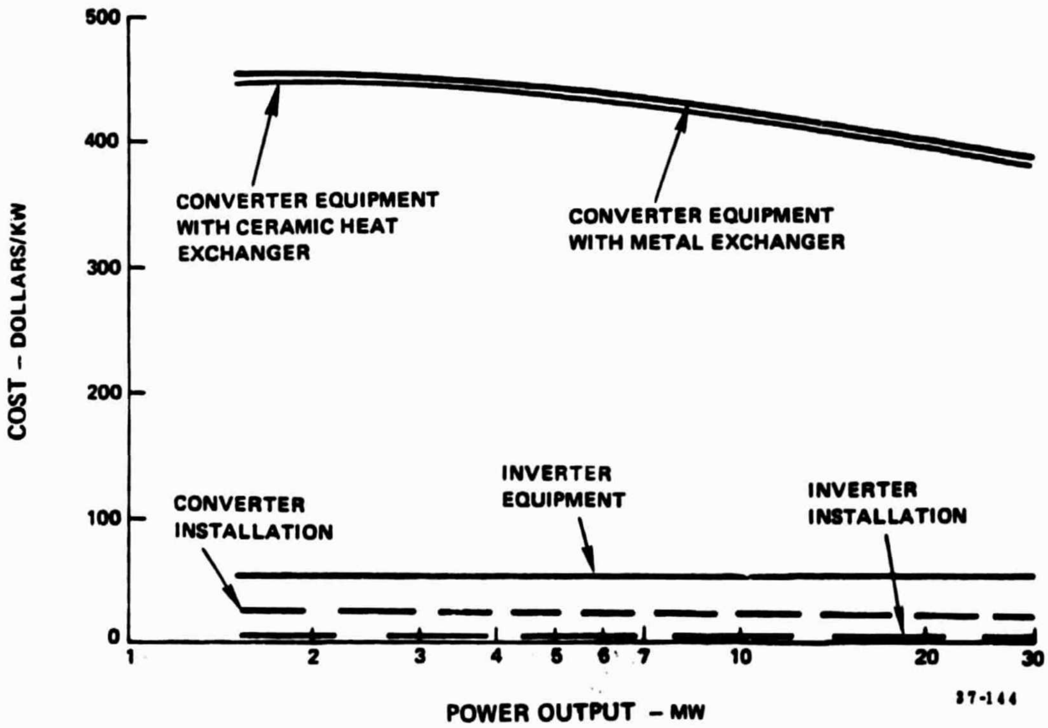

Figure III-176. Single Thermionic System Component Estimated - Costs

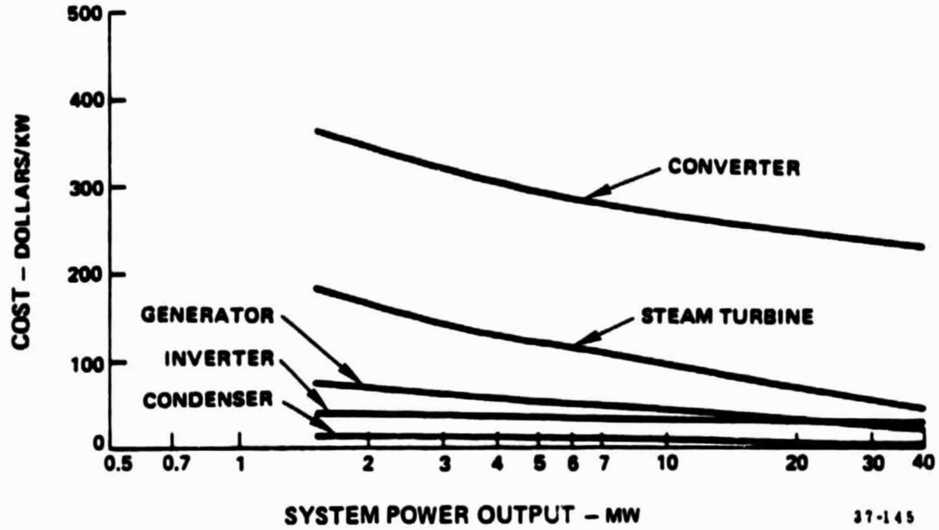

Figure III-177. Compound Thermionic System Component Equipment Estimated Costs - Ceramic Heat Exchanger

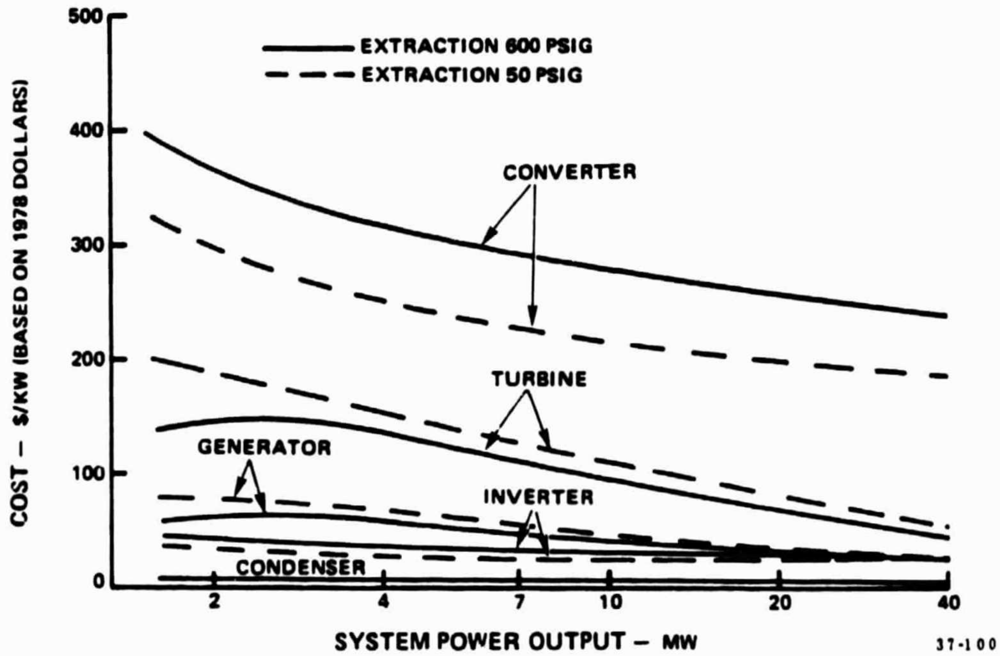

Figure III-178. Compound Thermionic System Component Equipment Estimated Costs - Metallic Heat Exchanger

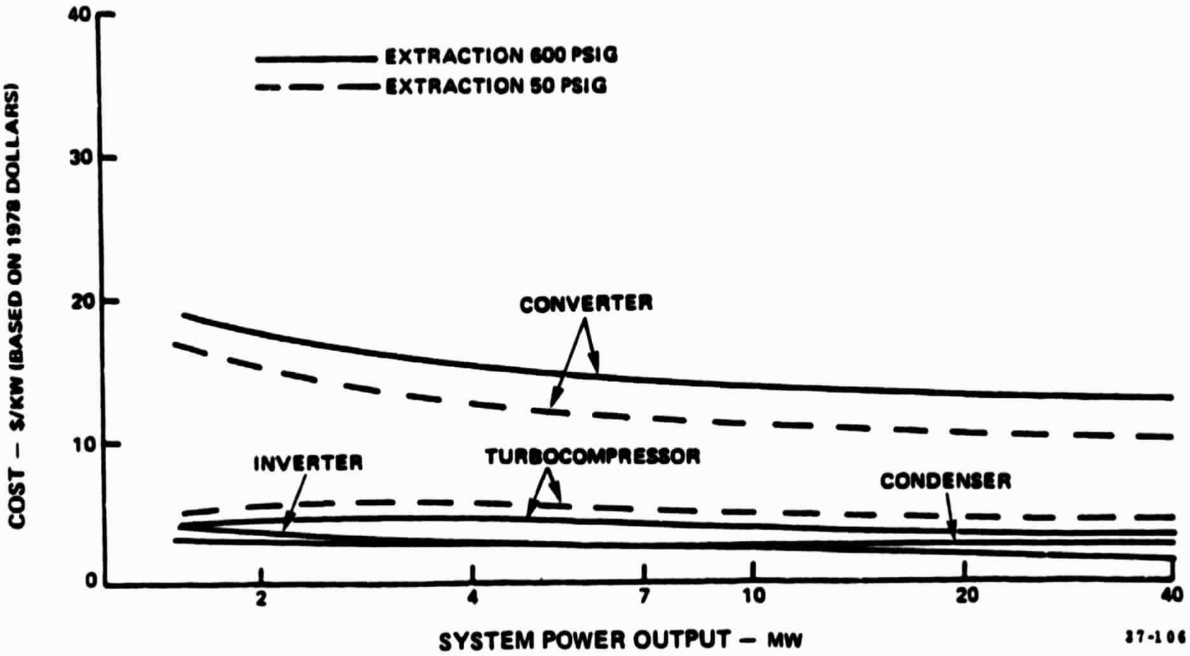

Figure III-179. Compound Thermionic System Estimated Installation Costs

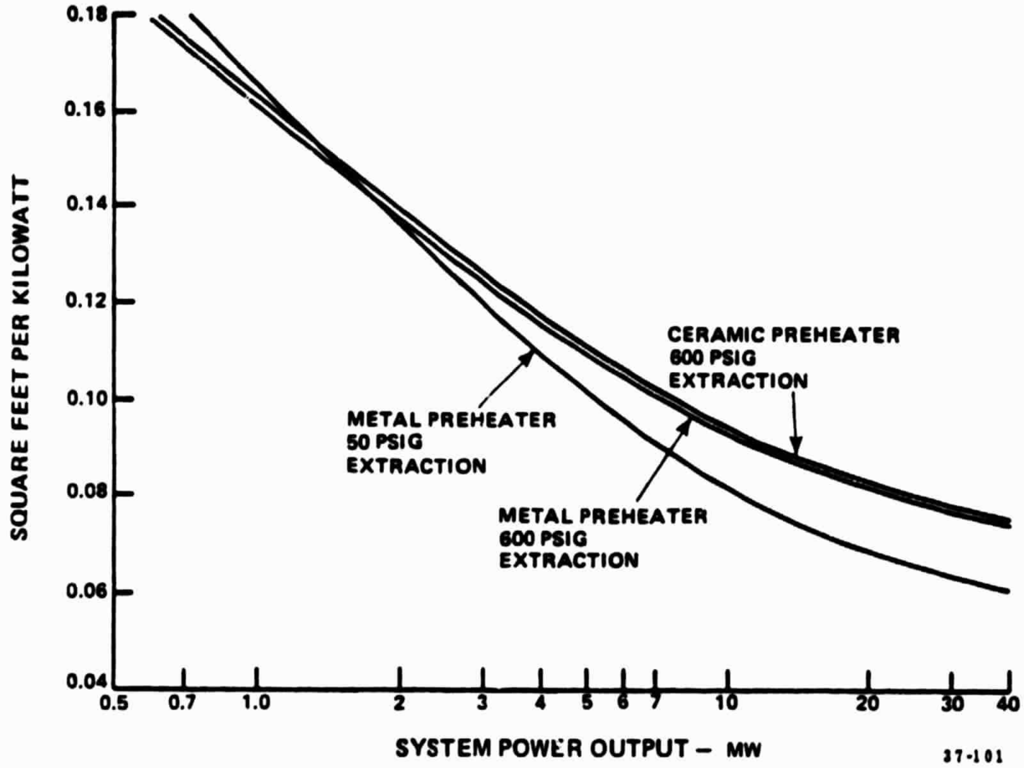

Figure III-180. Compound Thermionic Energy Conversion System Total Floor Area Required

MINORITY REPORTS

The Cogeneration Technology Alternatives Study required a broad scope of data and information energy conversion systems. The analysis also required objectivity and consistency. The conflicting needs of advocacy and objectivity were met by qualified experts for each energy conversion system with healthy self-interest in development and manufacture and by evaluators not involved in manufacturing using preset methodology. To encourage consistency, advocates for each technology offered comment and criticism at meetings where data for all energy conversion systems were reviewed. In addition, advocates could prepare "Minority Reports" as a means of comment or rebuttal.

The expertise and advocacy for thermionic energy conversion in this study were represented by Rasor Associates, Incorporated of Sunnyvale, California, who have prepared the following minority report.

SUPERIOR THERMIONIC COGENERATION SYSTEMS ALTERNATIVES NOT CONSIDERED IN COGENERATION TECHNOLOGY ALTERNATIVES STUDY

Introduction

The Cogeneration Technology Alternatives Study has performed an important function by identifying the energy savings which could result from the use of a large variety of cogeneration systems. In addition, it pointed out the possibility of designing completely new energy conversion system -- industrial process combinations (not included in the study) which could provide substantial additional energy savings. We believe this is particularly true in the case of thermionic energy conversion because of its unique properties. Several alternatives which might make use of the unique properties of thermionics were not examined in the study because of the ground rules. In particular, as a result of our experience as participants in the study, new thermionic cogeneration systems, conceived after the study was completed, have been identified which we of Rasor Associates

believe could save substantial additional energy annually. These savings are over and above those savings resulting from the conversion systems included in the study. The newly identified systems, described briefly herein, use various combinations of the following features:

1. The use of high temperature thermionic reject heat for direct thermal process requirements (particularly in the ethylene, steel, cement, glass and petroleum industries).

2. The use of high temperature thermionic reject heat for furnace air preheat.

3. The use of pulverized coal burning furnace.

4. The use of back pressure steam turbines instead of partial extraction turbines in thermionic-steam compound cases.

In order to maintain overall internal numerical consistency within the study, we cannot quantitatively describe the additional energy savings which would result from these concepts. However, a preliminary analysis performed by Rasor Associates without input from other study participants indicates that the savings are substantial. We believe that examination of these concepts could make thermionics one of the most attractive advanced energy conversion system options from a fuel energy savings point of view.

Thermionic Reject Heat for Direct Process Use

There are a number of industries which make use of high temperature direct process heat for manufacturing. Because of its very high heat rejection temperatures, thermionics may be particularly appropriate for use in such applications. Evaluation of methods to supply high temperature direct heat is important since high temperature direct heat needs constitute over 60 percent of the total 1990 projected energy requirement for the industries included in the study. Some of those direct heat needs is shown in Table III-63. Unfortunately, due to United

Technologies ground rules concerning cleanliness of direct heat and overall study ground rules which prohibited redesigning industrial processes to take advantage of the unique properties of each energy conversion system, the study was not able to evaluate the use of thermionic conversion to supply high temperature reject heat for manufacturing needs.

TABLE III-63. INDUSTRIAL THERMAL REQUIREMENTS INVOLVING DIRECT HEAT

Industry	Annual Fuel Needs 1990-Trillion Btu	Thermal Requirements (Million Btu/Unit) HW	300°F	500°F	700°F	Direct
Ethylene	4,512	0	0	0	6.9	50
Glass	213	0	0	0	0	8.6
Cement	694	0	0	0	0	5.3
Steel	4,533	0	0	1.2	0	13.2
Alumina	600	0	0	4.8	0	3.4
Rubber	48	0	1.3	3.4	0	1.6
Styrene	351	0	23.8	0	0	4.5
Petroleum	4,088	0	0	0.1	0	0.4
Gray Iron	154	0	0.06	0	0	6.5

Note: HW is hot water: 300, 500 and 700°F refer to the temperature of the required process steam; direct heat is heat supplied at or above 1000°F, usually in the form of a gas.

Thermionic Air Preheat

A universal cogeneration system which may be useable in almost any industry can be designed which has an energy savings ratio of nearly 0.2, i.e., an energy savings of 20 percent in a match T configuration compared to non-cogeneration. This is accomplished by using a thermionic burner. As shown in Figure III-181, combustion air for the burner is preheated by the heat rejected by the thermionic devices (up to 1200°F heat rejection temperature). The high temperature burner gases (at or above 2600°F) are used to heat the thermionic converters and provide direct heat for the industrial process. This energy conversion system whose

output is electric power and hot flue gas can be tailored to meet almost any industrial process heat requirement. The key to the design is the high heat rejection temperature of the thermionic converters.

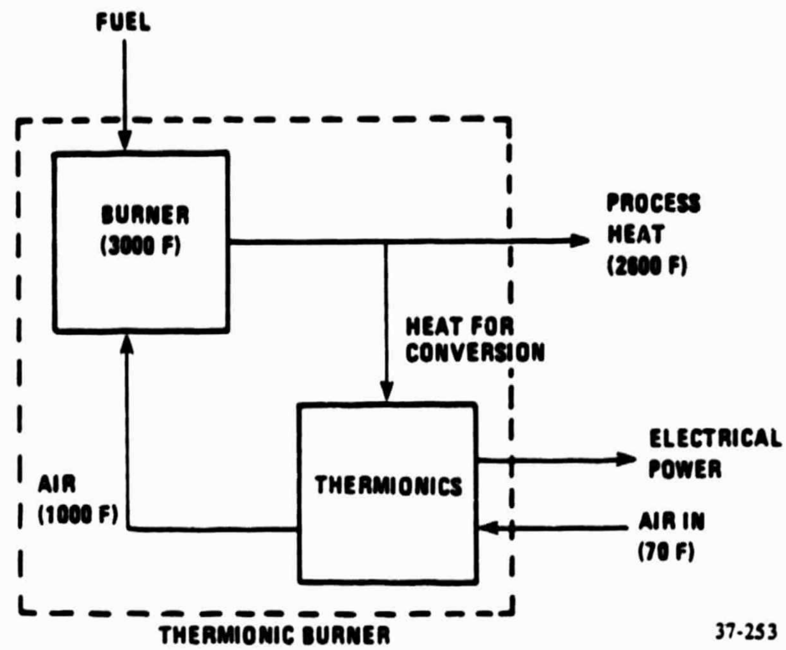

Figure 111-181. Energy and Gas Flow Within a Thermionic Burner

Direct Use of Coal

The United Technologies study only considered thermionic systems which use liquid coal-derived fuel, yet, like the steam turbine system, thermionics can be used in conventional coal-fired furnaces. A system using coal with its higher capital costs but lower fuel costs may be an attractive option and should be evaluated.

Thermionic Steam Compound Using Back Pressure Turbine

In evaluating the thermionic steam compound system, extraction turbines were used in the study. We at Rasor Associates, based on preliminary calculations, believe that the use of back pressure turbines could considerably increase the energy efficiency of the thermionic steam compound systems.

Conclusion

The study predicted energy savings for thermionic cogeneration of one and one half to two and one half quads per year. By modifying the study ground rules to take advantage of the unique characteristics of thermionics, Rasor Associates believes that very substantial additional savings can be predicted. We believe that based on the concepts identified above continued studies of advance cogeneration thermionic systems and the development of thermionic power modules for cogeneration use are fully justified.

ORGANIC RANKINE CYCLE

INTRODUCTION

The steam Rankine cycle has long been established as a power generating system. In more recent years, with development of many new organic fluids, the organic Rankine cycle has demonstrated its capability as a power system. The superiority of steam or organics as a working fluid is very much a function of the application and operating conditions. In the case of the industrial plant waste heat recovery system, the lower temperature operating conditions indicate a substantial advantage for the organic system. The major components of a closed loop Organic Rankine Power System for converting waste heat to electrical power are:

a. Waste heat boiler to convert the working fluid from liquid to vapor.

b. Turbine-generator to convert energy from the vapor expansion to electrical power.

c. Condenser to condense the expanded vapor to liquid.

d. Pump to return the working fluid to boiler operating pressure.

e. Heat rejection system to remove waste heat from condenser.

CONVERSION SYSTEM DESCRIPTION

The Organic Rankine power plant is essentially the same as the conventional steam Rankine power plant with the major exception of the working fluids; an organic compound vs. water. The characteristics of the organic working fluid have proved to be particularly desirable for use in Rankine power plants utilizing waste heat at moderate temperatures.

Figure III-182 is a flow diagram of the system showing the major components. The waste heat exhaust is ducted through the waste heat boiler which is designed to recover sensible heat from the exhaust gas at 800-1000°F to a boiler of approximately 300°F. The boiler exhaust temperature is limited to prevent any condensation in the boiler exhaust system. The waste heat boiler heats the organic fluid from a liquid at 150°F to a superheated vapor at 750 psia and 700°F. The superheated vapor drives the turbine generator to produce electric power. The turbine exhaust is condensed and drained to the hold-up tank for in-process storage. The boiler feed pump returns the working fluid to the boiler at 750 psia to complete the cycle.

The heat from the condenser provides hot water or is rejected to the atmosphere by a cooling tower.

System auxiliaries include a secondary condenser to condense the turbine bypass flow. Turbine bypass is used as a control function whenever the boiler output exceeds the turbine requirements. It is also used for startup and in the event of an emergency shutdown. A vacuum pump is used to exhaust noncondensables from the condenser. The vacuum pump output is condensed and filtered to remove any working fluid prior to exhausting to atmosphere. The power plant controls are similar to those of a steam power plant of similar size and designed for the specific requirements of the application. The controlled parameters are boiler pressure, boiler outlet temperature, working fluid flow, turbine speed, condensing pressure, liquid level and generated voltage. Indicating lights, alarms, and automatic shutdown devices are provided for safety.

PERFORMANCE

The performance of the following three bottoming application cases were studied.

1. Glass container production involving a high temperature (800°F - 1100°F) bottoming cycle in which the output is electrical power and hot water.

2. Cement production involving a high temperature (800°F - 1100°F) bottoming cycle in which the output is electrical power.

3. Cement production involving a moderate temperature (500°F - 700°F) bottoming cycle in which the output is electrical power.

The range of parameters studied is summarized in Table III-64. The minimum, maximum and design conditions resulted from industry surveys of the cement and glass industry. The working fluid selected for all cases is an azeotropic mixture of 2-methyl pyridine and water. This fluid was chosen for this temperature range based on its thermal stability, low cost, safety and high performance characteristics. A temperature entropy diagram for this mixture is presented in Figure III-183.

In general, two basic configurations are being evaluated. For the glass plant, the service hot water (140°F) will be produced from heat rejected in the power plant condenser. This will result in a higher condensing temperature of 150°F. For the other case, (two cement plants) with 1000°F and 700°F waste heat, a wet cooling tower will be used to provide condenser heat rejection. This configuration will produce a condensing temperature of 110°F. Note that cooling towers are included with the balance-of-plant in Volume IV of this report. Cooling towers for the organic Rankine cycle were included in the conversion system design for completeness.

The range of parameters selected reflect the technologies which could be available in the 1985-2000 time period, with 1993 considered to be the median year.

The power plant design is based on the assumption that major components of the system - the boiler, the turbine, and the condenser will all be upgraded to a higher level of technology than exists commercially today. The improvements in design will improve performance and reduce the size of the system. These improvements will be accomplished as a result of several generations of redesign based on system and component operating experience. There is currently little

operating experience with large organic Rankine cycle systems, therefore the first generation of equipment will be comprised of new designs and adaptations of commercial steam systems. Later designs will be upgraded as a result of operating experience and technological advancements.

The Rankine cycle power plant will use an advanced once-through boiler using new design and control technology to avoid tube burnout, tube fouling, and unstable operation. The turbine will be specifically designed for this working fluid and will be smaller in size and number of stages than an equivalent power level steam turbine. Condensing of the turbine exhaust will be in a water cooled heat exchanger. The boiler feed water pump and all other auxiliaries in the system will be driven by electrical motors. These auxiliaries represent a parasitic load on the total electrical generator output.

The energy system performance is shown in Table III-65. Performance is calculated for the nominal size plant and for plants representing the maximum and minimum ratings of the expected size range. It would be expected that the power plant rating would be selected to provide rated output under most process operating conditions. When the waste heat supply exceeds power plant rating either excess gas will be bypassed around the boiler or working fluid vapor will be bypassed around the turbine limiting electrical output to rated capacity. The minimum operating limit depends upon the process characteristics and should be established such that system fluctuations would never approach the minimum heat input necessary to maintain positive power generation.

Bottoming cycle energy utilization for the nominal ratings is shown on Figure III-184. Permissible range of operation for a typical power plant utilizing waste heat is shown on Figure III-185.

The performance of the energy conversion system as a function of the power plant size is shown on Figures III-186, III-187 and III-188.

COST ESTIMATES

The capital cost was obtained by sizing all major components and then using appropriate 1978 cost factors to obtain the cost of the major equipment. The installation costs are based on the use of the following factory assembled modules: (1) Boiler subsystem, (2) power conversion subsystem, (3) boiler feed and makeup subsystem and (4) heat rejection subsystem.

The modularized installation cost was taken as 125 percent of the equipment cost. This cost includes controls, piping, instrumentation, structures and module construction. The installation costs are based on retrofitting an existing facility and are expected to vary from 100 to 200 percent of equipment costs depending upon the particular site and the complexity of the installation. The value of 125 percent is used for an average installation.

A summary of capital cost of equipment and installation costs are shown in Table III-66. The costs as a function of plant size are expected to vary in accordance with a size exponent of 0.66 and are shown on Figure III-189.

The operating and maintenance costs were obtained from estimates of utility, labor, and replacement costs. The cost factor and assumptions are provided in Table III-67 along with the overall results. These results are based on the following criteria:

(1) 7200 hour/year plant operation
(2) Power plant automated
(3) Operator required for startup and shutdown only
(4) Major maintenance during annual plant shutdown
(5) Major equipment overhaul every 10 years

SCHEDULES

o Maintenance frequency

The maintenance goal for the design would be to have yearly major maintenance coinciding with the yearly maintenance of the cement and glass process equipment. This goal should be easily achievable for this type of power plant.

o Maintenance interval

Maintenance of the system should be possible in a two week period with a low level work force (2 men). Maintenance would include (1) heat exchange cleaning, (2) pump, motor and gearbox lubrication and oil change, (3) replacement of leaky controls, instruments and piping, (4) working fluid replacement and (5) inspection of turbine and generator.

o Time between overhauls

The need for major overhauls of the turbine, gearbox, generator, pumps, and heat exchanger would be scheduled for evaluation every 10 years. The design life of the system between major overhauls would be 25 years.

o Major overhaul interval

A major overhaul would take about one month with a moderate level of effort (5 men).

o Construction and installation time

After the system design is mature, the construction and installation time would be about 14 months for a normal site. This time would include component manufacture and assembly of modules in the factory facilities, subassembly checkout, cement plant site preparation and system installation and checkout. Actual system installation is scheduled to take about 4 months.

EMISSION, RESOURCE, AND IMPLEMENTATION FACTORS

The Rankine cycle power plants in general have very little in the way of waste streams for disposal. For all plants, once a year, the working fluid may be replaced with a fresh charge to maintain a high level of performance. For the cement plants, the waste heat boiler will collect about 10 percent of the solids in the waste stream. These solids, however, will be recycled back to the cement kiln feed; therefore very little will be actually wasted.

The power plant will run all electric auxiliaries off the plant turbogenerator set; therefore no external power is needed except for startup. Those plants which use wet cooling towers for the condenser water will require a water source for makeup water to the tower. This water is assumed to be provided by wells. Water is lost from the cooling tower by evaporation and spray to the atmosphere. In addition, a continuous "blowdown" of water from the cooling tower sump is normally needed to prevent scale buildup on the tower. This water would be directed to the storm drain or sewer.

The waste disposal and resource requirements are presented in Table III-68 for the plants at their nominal design points.

These Rankine cycle power plants would be designed as fully automatic plants requiring a minimum of operator attention and maintenance. They would be designed as modules which could be loaded on railroad flat cars for shipment to the sites. The modular construction is possible for all components and would minimize the on-site installation problems and cost. The power plant would be divided into a (1) waste heat boiler module, (2) a power package module and (3) a heat rejection and fluid storage module.

- Operational flexibility

The waste heat boiler would be installed at the exit of the cement kilns before the radiant heat exchanger and dust cyclones. A bypass arrangement would be used

to allow the cement kiln to operate when the power plant was off line. A bypass installation of the boiler in the glass plant would be used for the same reason. With this arrangement, the impact on the normal process operation will be minimal.

- o Retrofit potential to existing plants

Each cement plant is unique in the design of its gas cooling and dust collection system at the exit of the kiln. However, a modularized bypass boiler unit can be designed which can then be custom fitted to each plant. Therefore, the retrofitting potential to existing plant sites is very high. Cement plants with suspension preheaters at the kiln exit have a lower potential applicability both due to equipment arrangements and energy potential. The retrofit of glass plants would also be by a bypass arrangement but is more straight forward due to the relative simplicity of the waste gas exhaust system.

- o Retrofit potential for evolving technology advancements

As stated previously, the dry cement process with the suspension preheater (moderate temperature cement plant) has less retrofit potential because the waste heat conversion system will cost more and have a lower power output. Near term Rankine cycle technology advancements will not increase the power output at these low temperatures but could decrease the costs by modularized construction with low installation costs. Advances in hot gas control system would be advantageous for retrofit by hot gas bypass arrangements.

New technology projections in the glass industry are not expected to impact the retrofit potential for the waste heat recovery system.

- o Siting flexibility

The modularized system would be adaptable to all sites with a variation in the heat rejection system used. Preferable heat rejection would be via a wet cooling tower for cement plants and direct process water utilization for glass plants. If water is

not available, a secondary air cooled loop would be added to reject heat from the condenser cooling water. Direct air cooled condensers would not normally be used.

- o Potential reliability

Low life cycle system costs would be maintained by minimizing in field maintenance and modularized replacement of parts. The two areas where redundancy will be used to improve reliability are in (1) the boiler feed water pumps, and (2) key operational controls and instrumentation. Using these design principles, a high system reliability can be achieved.

SPACE REQUIREMENTS

The exact space requirements for the system will depend on each installation and are site specific, but general dimensions can be provided. For the cement plant, the waste heat boiler, turbogenerator set and condensers would all be elevated to the level of the kiln, about 40 feet off the ground. The holdup tanks, storage tanks, control room, and pump room would be located in the space below. The waste heat boiler would be located in hot gas bypass ducting. The turbine and generator equipment would be located in an elevated metal building, adjoining the boiler. For purposes of this study the glass plant can be assumed to fit in a similar envelope.

The wet cooling towers are included in the balance-of-plant presented in Volume IV of this report. They are included here for completeness and for the cement plants can be located at any convenient spot within about 150 feet from the power plant. The space requirements and dimensions are listed in Table III-69 for the three applications.

FUTURE DEVELOPMENTS

Future developments in organic Rankine bottoming systems will depend upon their successful usage in industrial applications. System and component improvements will be accomplished as a result of several generations of redesign and refinements based on system and component operating experience. There is currently little operating experience with large organic Rankine cycles systems; therefore the first generation of equipment will be comprised of adaptations of commercial steam systems. Later designs will be upgraded as a result of operating experience and technological advancements. Future developments in major components are assumed to proceed as follows:

(1) Turbines

Current steam turbines in the applicable power range provide efficiencies in the range of 50 to 75 percent depending on operating speed, power and number of stages. It is expected that first generation organic Rankine turbines will achieve 75-80 percent efficiency due to the better properties of the higher molecular weight fluids. Projected efficiency for the 1993 time period is 85% as a result of several generations of improved designs based on technological advancements and operating experience using organic working fluids. The organic turbine will also be between 25 and 50% of the size of equivalent power level steam turbines.

(2) Boilers

Waste heat boilers are currently available although their use in the industries selected for the specific cases has been quite limited with no experience in organic Rankine cycle systems. The improvements in boiler technology will provide smaller sizes and longer life as a result of improved heat transfer and better means of handling fouling from the high particulate content common to most waste heat gas streams. Currently available waste heat boilers are of the recirculation type requiring separator drums and downcomers or recirculation pumps. Improved control and fouling technology will allow the use of the once-through boiler with

its advantages of smaller size, reduced working fluid inventory and lower life cycle cost.

(3) Condensers

Condensers will also benefit from improved technology based on experience with organic Rankine systems resulting in smaller sized systems. The changes in this area are expected to be primarily refinements based on experience with organic fluids due to the current mature nature of heat exchanger technology. The use of vertical fluted tubes is an example of such a refinement.

Table III-64. Bottoming Cycle Case

Case	High Temperature 800 - 1100°F	High Temperature 800 - 1100°F	Moderate Temperature 500 - 700°F
Output	Electrical Power and Hot Water	Electrical Power	Electrical Power
Application	Glass Melting Furnace	Cement Kiln Without Preheater	Cement Kiln With Preheater
Temperature (Typical)	1000°F	1000°F	700°F
Heat Input Range, BTU/hr	5.8 to 58 x 10^6	11.6 to 86.7 x 10^6	6.2 to 46.5 x 10^6
Design Point, BTU/hr (1)	20.8 x 10^6	38.8 x 10^6	20.8 x 10^6
Waste Gas Exhaust Temp, °F	300°F	260°F (2)	260°F (2)
Waste Stream Composition, % Gas	89.7	92.2	92.2
Waste Stream Composition, % H_2O	10.3	3.8	3.8
Waste Stream Composition, % Solid	0	4.0	4.0
Heat Rejection	Hot Water	Wet Tower	Wet Tower
Condensing Temperature °F	150	110	110
Cooling Water Inlet Temp, °F	110	75	75
Cooling Water Outlet Temp, °F	140	100	100

Note 1. Heat in waste gas stream referred to 60°F.

Note 2. 260°F exhaust temperature based on no sulfur oxides in exhaust gas stream due to absorption by cement process.

Table III-65. Energy Conversion System Performance

	Glass 1000°F			Cement 1000°F			Cement 700°F		
	Design	Min.	Max.	Design	Min.	Max.	Design	Min.	Max.
Heat Source Capacity, BTU H	20.8 x 10^6	5.8 x 10^6	5.8 x 10^6	38.8 x 10^3	11.6 x 10^6	86.7 x 10^6	20.8 x 10^6	6.2 x 10^6	46.5 x 10^6
Heat Source Capacity, (Kw)	6092	1699	16990	11365	3398	25395	6092	1816	13620
Boiler Output (Kw)	3315	924	9244	7914	2366	17684	3477	1036	7774
Turbine Inlet Temp, °F	700	700	700	700	700	700	600	600	600
Turbine Inlet Pressure, psia	750	750	750	750	750	750	450	450	450
Turbine Eff., %	84	80	85	85	83	86	84	80	85
Turbine Shaft Power, Kw	864	229	2438	2196	641	4965	872	248	1972
Mech. Losses, Kw	17.3	4.6	48.8	44	12.8	99	17.4	5.0	39.4
Alternator Eff., %	95	92	96	96	93.3	96.3	95	92	96
Aux. Loads, Kw	23.9	6.7	66.6	129	38.6	288	50.7	15.1	113.3
Power Output Kw	780	200	2227	1937	549.5	4398	761	209	1742
Hot Water, Kw	2454	706	6843	NA	NA	NA	NA	NA	NA
Power Plant Eff., %	23.5	21.6	24.1	24.5	23.2	24.9	21.9	20.2	22.4
Net Elect. Eff., %	12.8	11.8	13.1	17.0	16.2	17.3	12.5	11.5	12.8
Heat Recovery Eff., %	53.1	53.3	53.4	17.0	16.2	17.3	12.5	11.5	12.8
Stack Loss %	45.6	45.6	45.6	30.4	30.4	30.4	42.9	42.9	42.9
Stack Loss Kw	2777			34[illegible]1			2615		

Table III-66. Waste Heat System Costs

ITEM			1000°F GLASS 780 KW		1000°F CEMENT 1,937 KW		700°F CEMENT 761 KW	
3.1	Primary Energy Converter							
	Turbine		56,200		114,600		56,200	
	Gearbox		9,800		20,000		9,800	
	Pumps		12,900		24,600		12,900	
	Misc. Mechanical		17,700		25,600		17,700	
	Tanks		10,700		18,500		10,700	
	Controls		20,800		25,000		20,800	
		Sub Total		128,100		228,300		128,100
3.2	Primary Generator			27,500		45,500		27,500
3.5	Bottoming Cycle Vapor Generator			115,200		198,600		198,600
3.6	Heat Recovery Equipment			38,800		NA		NA
3.7	Condensers			see 3.6		70,600		44,500
		Total Equipment Cost		309,600		545,000		398,700
		Total Equipment Cost ($/KW)		397		281		524
3.6	Assembly and Installation (1.25 x Equipment Cost)			387,000		681,000		498,370
	Assembly and Installation ($/KW)			496		352		655

Table III-67. Rankine Cycle Waste Heat Systems, Annual Operating Costs

		1000°F GLASS			1000°F CEMENT			700° CEMENT		
OPERATING COST	COST FACTOR	ITEM	UNITS	COST	ITEM	UNITS	COST	ITEM	UNITS	COST
Working Fluid Costs	$7.5/gal	2040	gal/yr	15,300	5200	gal/yr	39,000	2040	gal/hr	15,300
Manpower										
Operating Labor	$20/hr	150	Man/Hr	3,000	150	Man/Hr	3,000	150	Man/Hr	3,000
Maintenance Labor	$15/hr	1000	Man/Hr	15,000	1000	Man/Hr	15,000	1000	Man/Hr	15,000
Replacement Equipment	2% of Capital	309,600	$	6,200	545,000	$	10,900	398,700	$	8,000
Total Annual Operating Cost				39,500			67,900			41,300
Power/Year (KWH)				5.6×10^6			13.95×10^6			5.48×10^6
Operating & Maintenance Costs ($/KWH)				0.0070			0.0049			0.0075

Table III-68. Rankine Cycle Waste Heat Systems, Emissions and Resource Factors for Plants

	UNITS	GLASS PLANT	CEMENT PLANT	MODERATE TEMPERATURE CEMENT PLANT
Cooling Tower Blowdown	gpm	-	10	5
Cooling Tower Blowdown	$lb/10^6$ btu	-	116	108
Cooling Water Makeup	gpm	-	60	25
Cooling Water Makeup	$lb/10^6$ btu	-	696	540
Working Fluid Disposal	Gals/yr	2040	5200	2040
Working Fluid Disposal	$lb/10^6$ btu	0.09	0.12	0.09
Gas Stream Solid Waste[1] (collected in boiler hopper)	tons/yr	-	1765	1340
Gas Stream Solid Waste[1] (collected in boiler hopper)	$lb/10^6$ btu	-	10.83	15.34

[1]Solid waste will be recycled to cement kiln feed.

Table III-69. Rankine Cycle Waste Heat Systems, Space Requirements

ITEM NO.	PARAMETER	UNITS	GLASS PLANT	CEMENT PLANT	MODERATE TEMPERATURE CEMENT PLANT
1.0	Net Electrical Output	kw	780	1937	762
2.0	Power Module Width	ft	20	34	20
3.0	Power Module Length	ft	34	34	34
4.0	Power Module Height	ft	44	44	44
5.0	Power Module Ground Area (Item 2.0 x Item 3.0)	ft^2	680	1156	680
6.0	Power Module Volume (Item 5.0 x Item 4.0)	ft^3	29,920	50,864	29,920
7.0	Auxiliaries				
7.1	Wet Cooling Tower Width	ft	-	21	21
7.2	Wet Cooling Tower Length	ft	-	12	10
7.3	Wet Cooling Tower Height	ft	-	20	16
7.4	Wet Cooling Tower Ground Area	ft^2	-	252	210
7.5	Wet Cooling Tower Volume	ft^3	-	5040	3360
8.0	TOTALS				
8.1	Ground Area (Items 5.0 + 7.4)	ft^2	680	1408	890
8.2	Volume (Items 6.0 + 7.5)	ft^3	29,920	55,904	33,280

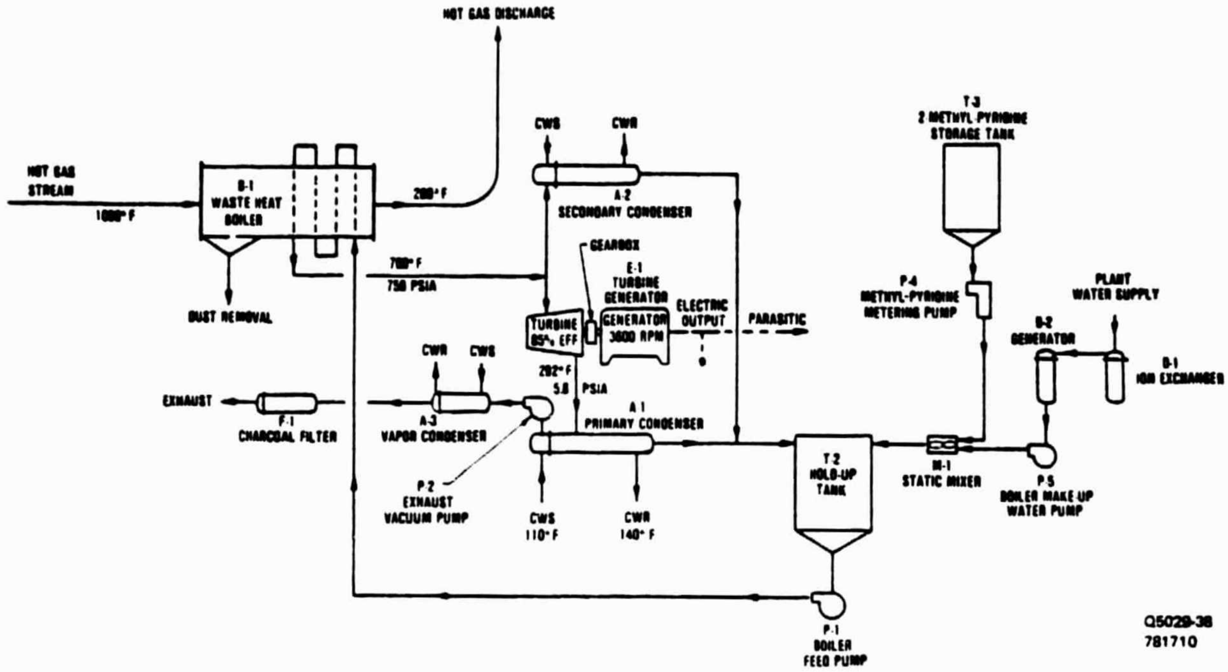

Figure III-182. Organic Cycle Waste Heat Recovery System Flow Diagram

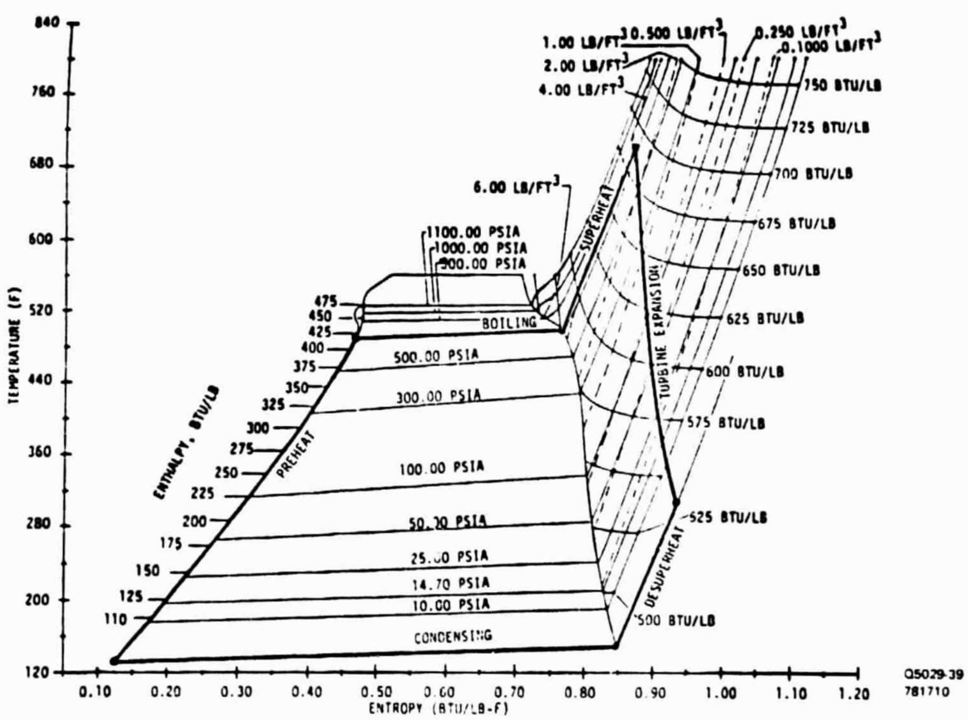

Figure III-183. 2 Methyl Pyridine/Water Temperature Entropy Diagram

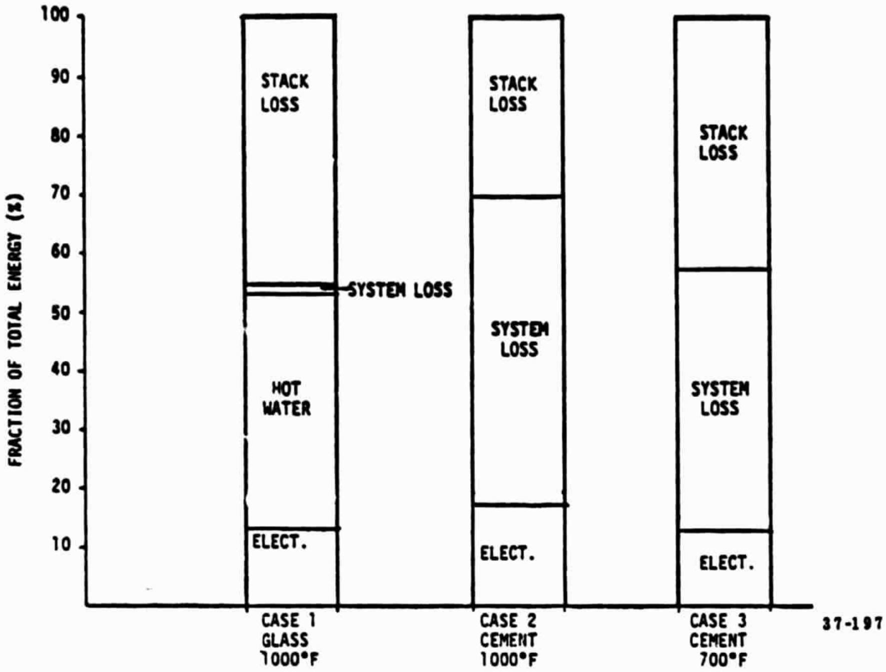

Figure III-184. Bottoming Cycle Energy Utilization

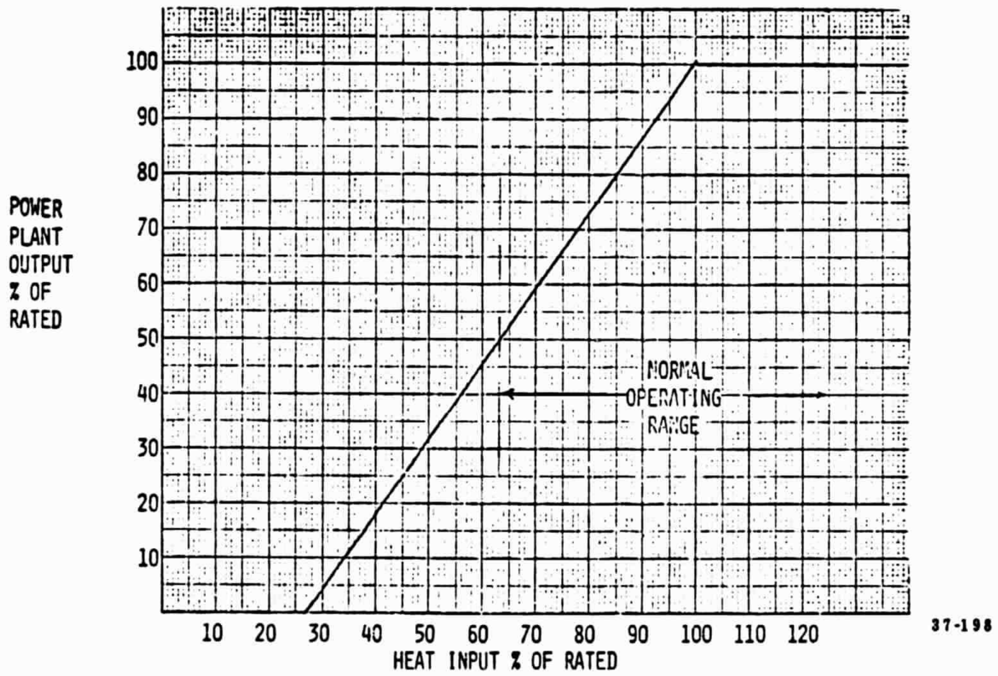

Figure III-185. Power Plant Output as Function of Heat Input

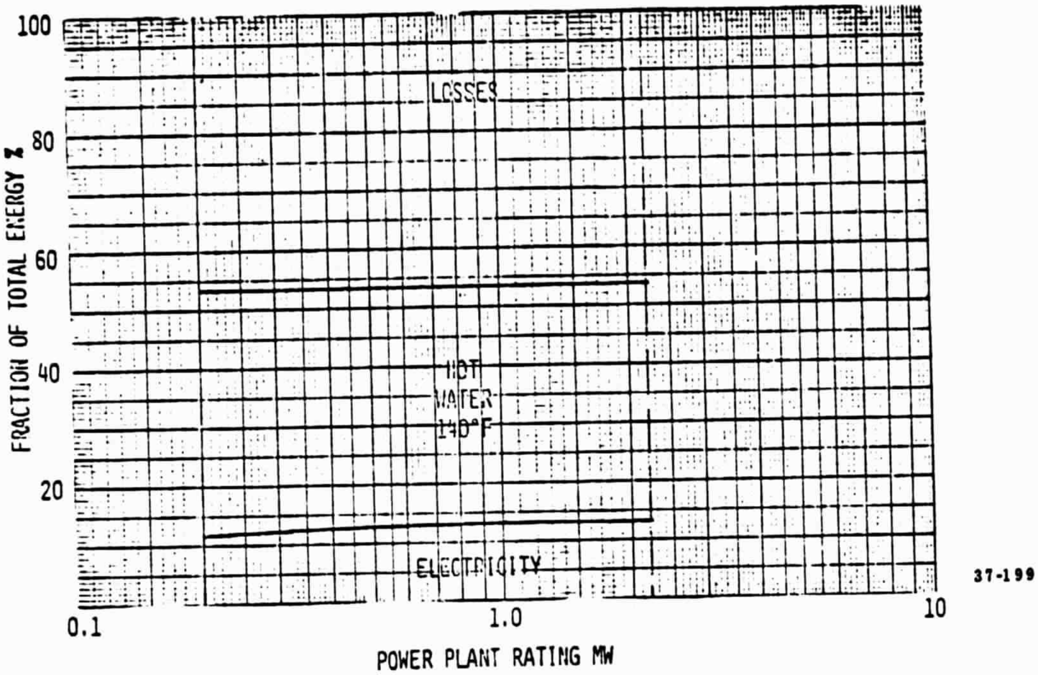

Figure III-186. Glass Melting Furnace Power Plant Performance

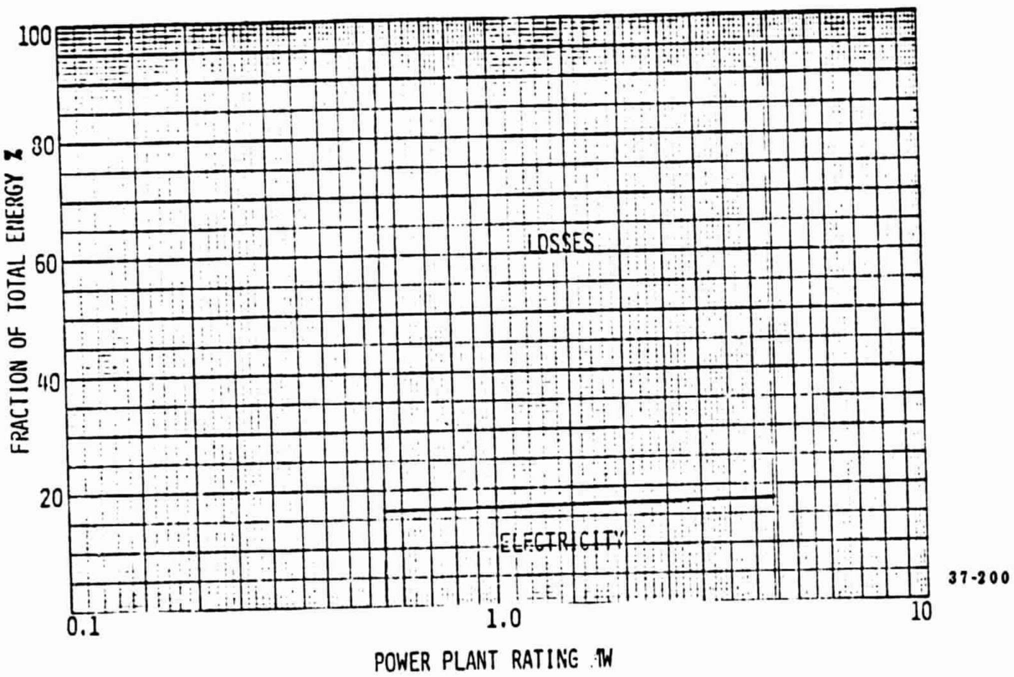

Figure III-187. Cement Plant (Without Preheater) Power Plant Performance

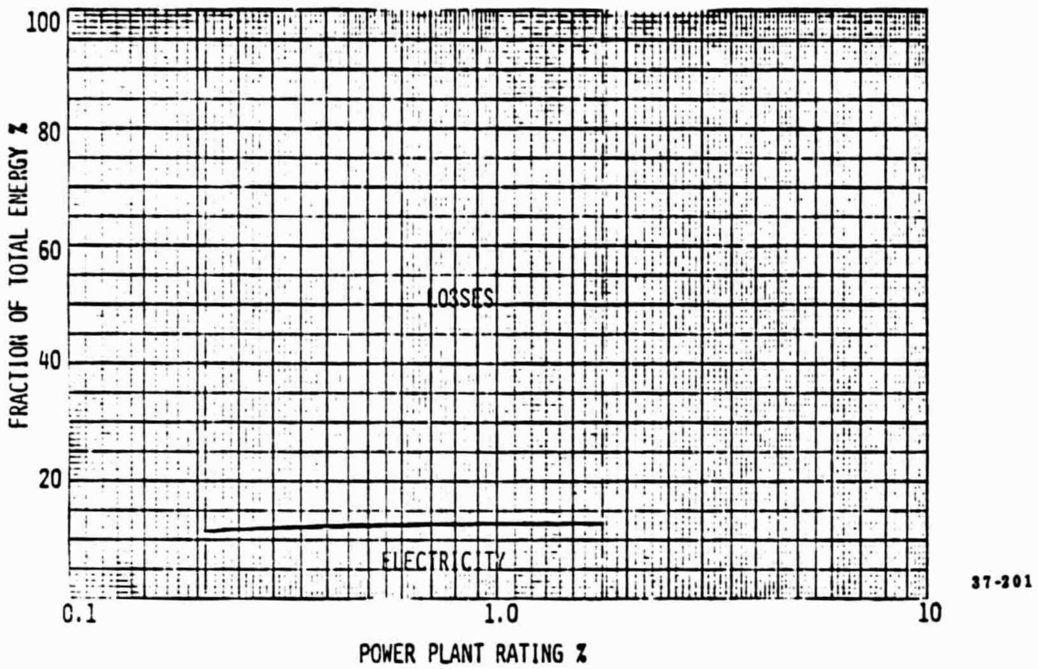

Figure III-188. Cement Plant (With Preheater) Power Plant Performance

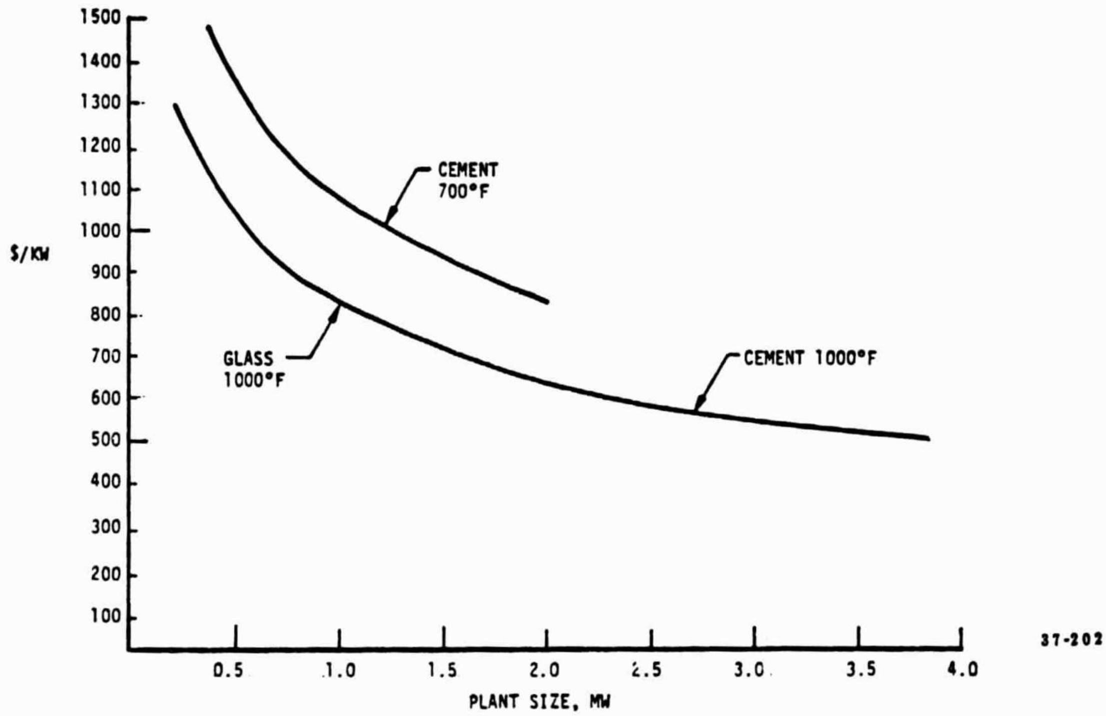

Figure III-189. Power Plant Instalation Costs vs. Plant Size